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DESIGN OF POWER-PLANT INSTALLATIONS

PRESSURE-LOSS CHARACTERISTICS OF DUCT COMPONENTS

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ADVANCE RESTRICTED REPORT

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SUMMARY

A correlation of what are believed to be the most reliable data available on duct components of aircraft power-plant installations is presented herein. The information is given in a convenient form and is offered as an aid in designing duct systems and, subject to certain qualifications, as a guide in estimating their performance.

The design and performance data include those for straight ducts; simple bends of square, circular, and elliptical cross section; compound bends; diverging and converging bends; vaned bends; diffusers; branch ducts; internal inlets; and angular placement of heat exchangers. Examples are included to illustrate methods of applying these data in analyzing duct systems.

INTRODUCTION

The objectives in the design of an aircraft duct system are to fit the components of the system within the available space and to meet an air-flow demand with a minimum of energy loss. Analyses of duct systems are, in general, made for one or more of the following purposes:

(1) Estimation of pressure loss in a duct

(2) Determination of rate at which air will flow through a given duct system

(3) Calculation of exit area required to obtain a desired rate of air flow through a given duct system

(4) Evaluation of airplane drag chargeable to flow through a duct system

Aircraft duct systems occur in an infinite diversity of forms but, for the purposes of design and analysis, must at present be treated as a series of component parts - such as bends, nozzles, and diffusers - for which design and performance data are available. Analyses of duct systems are generally step-by-step procedures in which changes in the energy and the physical state of the ducted air are followed progressively from the free stream ahead of the airplane through the successive duct components to the point of discharge from the airplane. Simplified procedures for making such analyses are given in references 1 and 2, and a precise, rigorous method is given in reference 3. These references are primarily concerned with analytical procedure and do not deal with loss characteristics of duct components.

A large amount of experimental data and some theoretical treatments of the flow in duct components exist, but the data often appear to be inconsistent and some of the theoretical treatments are contradictory. This lack of agreement is principally due to inadequate consideration of all variables affecting the flow characteristics - a natural consequence of the undeveloped state of the theory.

The purpose of this paper is to present, in simple and concise form, information useful for the analysis and design of duct systems for aircraft power-plant installations. Data are presented on design criterions and pressure-loss characteristics of straight ducts, duct bends of various cross-sectional shapes, vaned bends, branch ducts, and several types of diffuser. Several examples are presented to show methods used in analyzing duct systems.

In the present report the most reliable data available have been used but some of these data are recognized as questionable. In cases in which data from different sources are inconsistent, the material presented is, as far as possible, a mean weighted by consideration of the conditions under which the results were obtained.

In cases in which data for a particular type of duct component have been obtainable from only one source and were therefore without adequate corroboration, these data have been presented for lack of better.

The flow characteristics of any duct component are considerably affected by variations in the nature of the upstream flow; for the data presented the type of flow is that generated by a long straight pipe. Because of this effect and the limitations on available data, the present discussion of flow coefficients for duct components is subject to extension and revision when more comprehensive data become available. If the pressure and velocity distributions of the flow at the inlet of a duct component are not uniform, the total-pressure loss through the component will be greater than would be predicted by use of the present data. Subject to these qualifications, the material presented is offered as a guide in designing duct systems and estimating their performance; however, for the attainment of best performance, complete systems should be refined by tests of airplane models in wind tunnels or tests of duct systems in which the air flow is induced by blowers.

SYMBOLS

A duct cross-sectional area, square feet

a velocity of sound, feet per second

C_L lift coefficient (L/qc)

c length of vane chord, feet

D hydraulic diameter, feet

$$\left(\frac{4 \times \text{Cross-sectional area of duct}}{\text{Perimeter of duct}}\right)$$

d diameter, feet

F_c compressibility factor $\left(1 + \frac{1}{4}M^2 + \frac{1}{40}M^4\right)$

f friction factor for straight ducts $\left(\frac{1}{4} \cdot \frac{\Delta H}{q} \frac{D}{l}\right)$

s gap or vane spacing, perpendicular distance between vane chords, feet

H total pressure, pounds per square foot

h height of duct (in case of bend, dimension in plane perpendicular to plane of bend), feet

K arbitrary constant

k_1 bend-loss coefficient $\left(\frac{\Delta H}{q}\right.$ of bend divided by $\frac{\Delta H}{q}$ of equivalent constant-area bend with identical inlet$\left.\right)$

k_2 total-pressure-loss coefficient of diffuser expressed as fraction of loss due to sudden expansion $\left[\frac{\Delta H}{q}\right.$ of diffuser divided by $\left.\left(1 - \frac{A_{di}}{A_{de}}\right)^2\right]$

L lift, pounds per foot of span

l axial length of duct, feet

M Mach number (V/a)

m mass rate of flow, slugs per second

n number of vanes in duct bend

P perimeter of duct cross section, feet

p static pressure, pounds per square foot

Q volume rate of flow, cubic feet per second

q dynamic pressure, pounds per square foot $\left(\frac{1}{2}\rho V^2\right)$

R Reynolds number $(\rho VD/\mu)$

r radius, feet

$\bar{r}$ mean radius of bend, feet $\left(\frac{r_a + r_b}{2}\right)$

T temperature, °F absolute

V velocity in duct, feet per second

V_0 free-stream velocity, feet per second

w duct width (in case of bend, dimension in plane of bend), feet

x,y abscissa and ordinate of standard coordinate system

α angle of attack in relation to air-stream direction, degrees

β angle of duct bend, degrees

λ angle of junction of duct and resistance unit, degrees

ρ density of air, slugs per cubic foot

μ absolute viscosity of air, pound-seconds per square foot

ΔH total-pressure loss, pounds per square foot

ΔH_λ total-pressure loss due to angle between duct and resistance unit

Δp static-pressure loss, pounds per square foot

ΔT change in temperature, $^{\circ}F$

ΔV total vector-velocity change, feet per second

θ one-half equivalent conical angle of expansion, degrees

φ one-half angle between straight walls of partially curved diffuser, degrees

$\Delta H/q$ total-pressure-loss coefficient

$\bar{r}/w$ radius ratio

h/w aspect ratio

Subscripts:

a inside wall of bend

b outside wall of bend

d diffuser

e exit

f face

fi flared inlet

i inlet

r resistance unit

x arbitrary station

0 in free stream

1,2,3,... stations in duct system

max maximum

min minimum

GENERAL PRINCIPLES OF DUCT DESIGN

Skin friction and flow separation are two fundamental causes of pressure loss in fully turbulent flow through any duct component. The loss in a given duct component from each of these causes is roughly proportional to the dynamic pressure of air flow. Since the dynamic pressure of the air flow is proportional to the square of the flow velocity, the first basic principle in the design of efficient ducts is the maintenance of a low flow velocity by the use of ducts of adequate size. The importance of this principle may be illustrated by noting that, for a given rate of air flow, halving the diameter of a circular duct multiplies the velocities by 4 and the losses by 16.

Although skin friction is the dominant cause of pressure loss in flow through straight ducts of constant cross section, this pressure loss is small compared with the losses that occur when the main flow separates from the duct walls and thus creates areas of reverse flow and violent turbulence between the main flow and the duct wall. These areas require velocities in the main stream higher than are otherwise necessary. The second basic principle in the design of efficient ducts, therefore, is the maximum reduction of flow separation.

One type of flow separation occurs when forces arise in the air stream in a direction opposite to the direction of flow. Such a force is the pressure rise (or "adverse pressure gradient") produced by a deceleration of the air flow - for example, the deceleration of the air flow in a diffuser. The rate of pressure rise that may occur without producing flow separation depends on the velocity of flow near the duct wall, because the presence of thick boundary layers of slow-moving air is conducive to separation. Conversely, a decreasing pressure in the direction of flow (or a "favorable pressure gradient"), such as occurs in a nozzle, tends to prevent separation.

Changes of flow direction, as in bends, also give rise to forces that tend to cause separation of flow from the inner surface of the bend. Surface roughness or protuberances that cause local disturbances or retardation of the air near the duct wall aggravate conditions of incipient separation. Screens or resistances across the entire duct, on the other hand, tend to stabilize the flow and oppose separation by resisting flow increases in the center of the duct at the expense of the flow near the walls of the duct.

PROPERTIES AND DESIGN OF DUCT COMPONENTS

Pressure-loss characteristics and design criterions of several typical duct components are given in figures 1 to 16. The total-pressure-loss coefficient $\Delta H/q$, a ratio of loss in total pressure to dynamic pressure at the entrance to the duct component, has been given directly wherever possible; in all other cases, coefficients are given from which the pressure-loss coefficient can be computed.

Straight ducts of uniform cross section.- The pressure-loss coefficient for straight ducts of uniform cross section is given by the relation

$$\frac{\Delta H}{q} = 4\frac{l}{D}f \qquad (1)$$

The friction factor f varies with the character of the duct surface and the Reynolds number based on mean air velocity and the hydraulic diameter of the duct. Values of f obtained from figure 51 of reference 4 are plotted against Reynolds number in figure 1. Data in figure 13 of reference 5 agree closely with values in figure 1. Determination of the Reynolds number is facilitated by supplementary curves obtained by plotting the ratio of mass rate of flow to duct perimeter against Reynolds number for a number of air temperatures. The kinetic viscosity of the air used in constructing the supplementary curves of figure 1 was determined by Sutherland's equation as presented in reference 6.

A typical value of $\Delta H/q$ for straight aircraft ducts is $0.02\frac{l}{D}$, which is usually inconsequential compared with other parts of the system, and the loss in sections of straight ducts is generally neglected. Long winding ducts of small diameters, such as cabin-heater ducts, are sometimes treated as straight ducts of higher than average pressure loss due to friction. The use of

$$\frac{\Delta H}{q} = 0.04\frac{l}{D}$$

is recommended in reference 7.

<u>90° bends of constant-area rectangular cross section</u>.- Pressure-loss coefficients of 90° bends of constant-area and rectangular cross section given in figure 2 for three values of Reynolds number based on hydraulic diameter are derived from data appearing in references 5 and 8 to 12. The beneficial effect of large radius ratio appears throughout the range of R but the optimum aspect ratio shows a marked change with Reynolds number.

<u>90° bends of constant-area elliptical cross section</u>.- Pressure-loss characteristics of 90° bends of constant-area elliptical cross section are given in figure 3 for three values of Reynolds number. The data include circular ducts as a special case and were derived from data in reference 5. The benefits of large radius ratio and the existence of an optimum aspect ratio are noted for the bends of constant-area elliptical cross section as well as for rectangular bends. The effects of Reynolds number are much less for bends of elliptical cross section than for bends of rectangular cross section and appear mainly for the bends of high radius ratio.

90° bends of changing area.- Significant data (derived from reference 11) concerned with the relation of area change to the loss in 90° bends of a particular geometry are shown in figure 4. In this figure the ratio of loss in a bend with changing area to that in a bend with identical inlet form but constant area is plotted against the ratio of entrance width to exit width of the nonuniform bend. Important reduction of loss in converging bends and serious increases in loss in diverging bends are noted; the loss increases are particularly serious for bends of small radius.

Simple bends other than 90°.- No satisfactory correlation has been made of data for variation of pressure-loss coefficient with angle of bend. Pressure loss of 45° bends can apparently vary from one-third to two-thirds the loss of a similar 90° bend, according to the test conditions.

Compound bends.- Pressure-loss coefficients for three types of compound bend (fig. 5) derived from reference 5 are shown in figure 6. Inasmuch as differences in the losses between the U-, Z-, and 90°-offset bends appear from reference 5 to be small and inconsistent, the curves presented are averages of results for the three types of bend. There appears to be little variation of loss with Reynolds number. Introduction of a 5-foot spacer between the two parts of the compound bend has relatively little effect on the over-all loss but tends to give higher values for optimum aspect ratio. A comparison of the 180°-bend (U-bend) data of figure 6 with the 90°-bend data of figure 2 shows that the relative loss varies to a marked degree with the radius ratio and aspect ratio of the bend.

Effects of surface roughness on bend losses.- The effect of surface roughness on the losses in straight pipes has already been given by the curves of figure 1. A study of pressure-loss data for bends of angles from 30° to 90° and radius ratios from 1 to 6 (reference 11) indicates that the influence of surface roughness on the loss in bends, and presumably of other duct components in which major flow disturbances arise, is very much greater than can be attributed to the increase in skin friction at the mean velocity of flow. Analysis of the data in reference 11 suggests that the ratio of losses through two bends, identical except for surface roughness, is equal to the 1.75 power of the ratio of friction factors; that is,

$$\frac{\left(\frac{\Delta H}{q}\right)_2}{\left(\frac{\Delta H}{q}\right)_1} = \left(\frac{f_2}{f_1}\right)^{1.75} \tag{2}$$

(The subscripts 1 and 2 in this equation are used to denote the two bends of different surface roughness.) The exponent greater than unity can be explained by the fact that any deviation from a uniform velocity distribution because of extensive boundary-layer separation or the existence of secondary flows would require that some of the flow be at velocities greater than the uniform velocity. Equation (2) would not, therefore, be expected to apply for a duct component not involving extensive secondary flows or separation.

Equation (2) can be used to correct the bend-loss data of this report to values corresponding approximately to flow through duct bends with rough surfaces. The total-pressure-loss coefficient for smooth-surface bends can be determined from the data curves of figures 2 to 4 and 6. The curves labeled "Smooth surface" in figure 1 are used to determine the friction factor for smooth-surface bends. A representative value of friction factor for rough surfaces corresponding to ducts in production airplanes with the usual manufacturing irregularities is 0.01.

Vaned bends.- Vanes may often be advantageously used in duct bends, especially when an unfavorable radius ratio or aspect ratio must be tolerated because of some limitation peculiar to the particular design. A correctly designed vane installation will improve the velocity distribution at the exit of the bend and will generally reduce the pressure losses through the bend. The reduction in pressure loss arises from the fact that the flow in a good vaned-turn installation approaches that flow which would occur if the passage were divided into smaller passages of the same depth but shorter width and, consequently, of more favorable aspect and radius ratios. When more than three vanes are used, practical considerations usually require a bend with evenly spaced vanes and equal inner and outer radii. The value that these radii may attain is usually limited by the space requirements. Figure 7 shows an installation of thin circular-arc vanes and defines the variables concerned in the design of such a vane installation. The vanes are equal in radius and chord to the curved portion of the duct surface. From figure 7 it can be seen that the chord c is equal to $2r \sin \frac{\beta}{2}$.

From material given in reference 11, the following expression for the number of vanes required can be derived:

$$n = \frac{2}{C_L} \frac{\Delta V}{V_1} \frac{w_1}{c} - 1$$

The quantity ΔV is the vector difference of the velocities upstream and downstream of the bend, as illustrated in figure 7. For a given bend configuration, therefore, the number of vanes depends on the lift coefficient at which the vanes are to operate. If too high a lift coefficient is assumed in determining the number of vanes required, high losses and a poor velocity distribution downstream of the bend will result. An assumed lift coefficient that is too low will result in too many vanes and the total-pressure loss through the bend will again be excessive. Reference 9 indicates that, for thin vanes installed in a 90° bend, use of a lift coefficient of 0.8 gives approximately minimum losses and a satisfactory velocity distribution. It is not known whether $C_L = 0.8$ is the optimum for thin circular-arc vanes for bend angles other than 90°, but a study of reference 13 indicates that use of this value in designing bends other than 90° bends should give satisfactory results. Results given in reference 9 show that for a 90° bend the angle of attack of the vanes α should be 48°, or 3° more than half the angle of bend. For other angles of bend, the amount by which the angle of attack exceeds half the angle of bend might be adjusted proportionately to the angle of bend as a first approximation; that is, for a 45° bend, an angle of attack of 24° would be indicated.

For a 90° bend with inlet and outlet the same in area and shape, equation (1) reduces to

$$n = \frac{2}{C_L} \frac{w}{r} - 1 \qquad (3)$$

By using the value of $C_L = 0.8$ for thin vanes, equation (3) becomes

$$n = \frac{2.5}{r/w} - 1$$

Results for vanes which have two different thickness distributions applied to mean lines approaching a circular arc are given in reference 9 and show that, for the optimum vane installation, the loss coefficient $\Delta H/q$ is about 0.25, a value relatively insensitive to vane thickness. For vane installations other than the optimum, the losses are higher and vary considerably with the profile of the vane. The angle of attack for thick vanes is approximately the same as for the thin circular-arc vanes and small variations from the optimum angle of attack do not appreciably affect the pressure loss. Values of C_L from 0.9 to 1.0 may be used in determining the optimum number of these vanes to be used.

Thin vanes of noncircular profile, which are suitable for installation in bends of equal inlet and exit cross-sectional areas, have been developed theoretically by Kröber (references 9, 10, 13, and 14). Profiles for these vanes are given in table I and figure 8(a). Tests (reference 13) indicated that installations using a vane of the type developed by Kröber are very efficient, as shown by the low losses given in figure 8(b). The required number of vanes for a given installation can be determined directly from the chord length and the gap-chord curve of figure 8(b). The break in this curve between angles of bend from 45° to 60° is apparently a result of the methods used in developing the profiles. References 9, 13, and 14 give specific data only for angles of bend of 30°, 45°, 60°, and 90°.

Diffusers.- Losses of straight-wall diffusers of circular cross section may be computed from the curve of figure 9, which was derived from figure 10 of reference 15 and figure 1 of reference 16. The loss coefficient is given by the relation

$$\frac{\Delta H}{q} = k_2\left(1 - \frac{A_{di}}{A_{de}}\right)^2 \qquad (4)$$

where k_2 is the quantity plotted in figure 9 against the equivalent conical angle of expansion. The loss due to an abrupt expansion is obtained from equation (4) by taking k_2 equal to unity. To a limited extent, the losses of diffusers of noncircular cross section, particularly those of square cross section, are approximated by the loss of an "equivalent conical diffuser" which has a circular cross section and of which the length, the inlet area, and the outlet area are equal to those of the noncircular diffuser.

The most efficient straight-wall diffusers are shown in figure 9 to be those of equivalent conical angles of expansion between 3° and 10°. Frequently, however, because of restrictions on the length of diffuser, it is necessary to diffuse at angles higher than 10°. Curved-wall diffusers (references 14 and 15), such as the design shown in figure 10, have been found to have appreciably higher efficiencies than straight-wall diffusers, especially at high angles of expansion. The performance for this type of diffuser is also shown in figure 10. At the higher angles of expansion, the lower pressure losses are obtained by diffusing gradually in the first part of the diffuser and more abruptly in the last part in order to delay the separation point in the flow. Tests reported in reference 15 show no gain when the angle 2φ is made greater than 40°. Other sources (unpublished) indicate that, if the angle 2φ is greater than 60°, large losses will occur.

Diffusers followed by resistance units, such as intercoolers, are subject to lower pressure losses at high angles of expansion than are indicated in figure 9. An experimental investigation to determine the shapes of circular diffusers for highest diffuser efficiencies in diffuser-resistance combinations is reported in reference 17. Figure 11 is a sketch of the optimum shape and a plot of the included angle between the straight walls of the diffuser 2φ against the equivalent conical angle of expansion 2θ. The values of 2φ are those values that gave the highest diffuser efficiency. The solid and long-dash curves of figure 12 show the pressure losses in terms of the loss due to sudden expansion for diffusers designed according to figure 11. The short-dash curve of figure 12, which is an extension of the curve given in figure 9, applies to straight-wall circular diffusers not followed by resistance and is shown for comparison.

Branch ducts.- The problem of taking branches from a main air duct resolves into division of the main air stream and diversion of one or more of the consequent subdivisions of the main stream. Division should be made as nearly as possible on a basis of relative air flows and is best accomplished with dividers or splitters of rather blunt-nose airfoil shape, such as the NACA 0021 airfoil section. (See fig. 13.) Enlargement of cross sections immediately downstream of the point of division and in bends is to be avoided. Entrances to branch ducts should be normal to the air flow. Figure 13 illustrates the application of these principles and shows the division of the main stream, the diversion of one stream, and the subsequent subdivision of the diverted stream.

The internal-duct inlet is a special problem associated with branch ducts. The inlet of a duct that taps air from a chamber in which the air is essentially stagnant is known as an internal inlet. Figure 14 shows several examples of such inlets with accompanying representative values of pressure-loss coefficient taken from reference 11. The designs subject to the least pressure losses are the flared entrances, particularly the design using a lemniscate. The equation of the curve in polar coordinates is

$$r^2 = 2K^2 \cos 2\theta$$

The part of the lemniscate used in the inlet design extends over a range of θ from 16° to 45° (fig. 14).

Flow-resistance units set at angle to upstream duct.- The meeting at an angle of the incoming air with the face of a resistance unit causes a total-pressure loss that depends on the amount of angle, the efficiency of the resistance-unit core in its action as a turning vane, and the air-stream velocity. Data on these losses, from which the curves of figure 15 were derived, were obtained from reference 18 and from the Wright Aeronautical Corporation and the Naval Aircraft Factory. The data apply to intercoolers, circular oil coolers, and a viscous-impingement type of air filter. The geometry of the ducts and resistances is also shown in figure 15. The curves indicate that the pressure loss is similar to the pressure loss of a duct bend in that the aspect ratio of the resistance-unit air passages is a controlling factor.

ILLUSTRATIVE EXAMPLES OF DUCT ANALYSIS

Several examples illustrating the calculation of pressure loss, air flow, exit area, and internal drag for duct systems I and IV of figure 16 are given in tables II to IV. Each of the hypothetical duct systems shown in figure 16 adheres to the same general space requirements and has an over-all increase in the cross-sectional area from ~~0.5~~ .25 square foot at station 1 to 3.0 square feet at station 6. The selection of the pressure-loss coefficients is illustrated for system I in table II. Step-by-step computations for systems I and IV are given in tables III and IV, and the pressure-loss distributions of the four systems are compared in figure 17. $A_1 = .25\ ft^2$

Duct system I (fig. 16) was designed according to the two basic principles of duct design set forth in the section entitled "General Principles of Duct Design." The high-velocity air at station 1 is expanded in a diffuser having an equivalent conical angle of expansion of 7°, which is shown in figure 9 to be subject to minimum pressure losses. The diffuser is followed by a well-rounded 90° bend of constant cross-sectional area. The rest of the diffusion is accomplished at a higher rate in a diffuser having an equivalent conical angle of 13.8°. Although the rate of expansion is high in the second diffuser, the loss is not excessive because of the low dynamic pressure at the entrance. The second 90° turn is quite sharp but does not cause a large pressure loss because of the low-velocity air. Duct system II (fig.16) was designed so that part of the area expansion is accomplished in the first 90° bend. Duct system III is an example of a compromise which emphasizes more than system I the principle of having low flow velocities. The low flow velocity is obtained by diffusing at a higher rate of expansion. Duct systems III and IV represent opposite extremes in relation to the initial expansion of the air. In system III the expansion is accomplished rapidly in a diffuser having an equivalent conical angle of 16° located upstream of the first bend; in system IV all the expansion is accomplished between the two 90° turns with the area constant from stations 1 to 3.

The duct systems were assumed to be installations in an airplane flying at sea level in Army summer air at a true airspeed of 400 miles per hour. For simplicity, the total-pressure losses from the free stream to station 1 were assumed to equal the pressure rise given the air by the propeller; therefore, the total pressure at station 1 is equal to the free-stream total pressure. The adiabatic temperature rise from the free stream to station 1 was calculated by use of the following equation from reference 2:

$$\Delta T_{0,1} = 0.832\left[\left(\frac{V_0}{100}\right)^2 - \left(\frac{V_1}{100}\right)^2\right] \tag{5}$$

The total-pressure loss through each duct unit was calculated from the curves of this report as illustrated in table II for system I. The compressibility correction to the dynamic pressure was neglected except at stations 0 and 1 because of the low velocities. The following equation (from reference 19) was used to calculate the compressibility factor F_c at stations 0 and 1:

$$F_c = 1 + \frac{1}{4}\left(\frac{V_x}{a}\right)^2 + \frac{1}{40}\left(\frac{V_x}{a}\right)^4$$

The temperature from stations 1 to 5 was assumed constant because the systems contained no heat exchangers and the static-pressure changes were insufficient to cause significant changes in temperature. With the foregoing conditions and assumptions, the properties of the air at each station were calculated as shown in tables III and IV.

The total-pressure losses for each system are plotted against the duct stations in figure 17, in which system I is shown to be the most efficient. The high losses associated with bends of increasing cross-sectional areas are verified by the curve for system II. The curve for system III emphasizes the importance of efficiently diffusing the high-velocity air even at the expense of greater bend losses, providing the bend design is reasonably good. The data for system IV indicate the importance of efficiently reducing the air velocity as soon as possible even in those cases in which the efficiency of some of the following units must be reduced.

The calculations for system I have been extended to illustrate the method of obtaining air flow, exit area, and internal drag. Because the calculation of pressure drops across heat exchangers is a problem outside the scope of this report, the heat-exchanger pressure drop is not considered in the subsequent discussion. The nature of the calculation is in no way affected by this simplification, but the resultant drag, internal-drag power, and exit area will consequently be much too small to be representative. A well-designed exit duct was assumed to extend from station 6 to station 7, the exit, and the total-pressure losses in this contracting section were assumed to be negligible. Several mass air flows through the system were assumed and the estimated total-pressure losses, exit velocity, exit area, and internal-drag horsepower were evaluated for each air flow. The static pressure at the exit was assumed to equal the static pressure of the free stream; the temperature drop associated with the drop in static pressure from station 6 to the exit at station 7 was assumed to be adiabatic. The following equation expresses this adiabatic relation:

$$T_6 - T_7 = \Delta T_e$$
$$= T_6 x \left[1 - \left(\frac{p_7}{p_6}\right)^{0.286}\right]$$

The exit velocity V_7 was calculated by substituting ΔT_e and V_6 in equation (5). The calculations for a mass air flow of 0.109 slug per second are summarized in table III. The internal-drag horsepower caused by the momentum deficiency of the discharged air and the exit areas required to obtain certain mass flows through the system are plotted against mass air flow in figure 18. From these curves the exit area required for a given mass flow or, conversely, the mass flow corresponding to a given exit area, may be determined. If a heat exchanger had been included in the foregoing arrangement, the pressure drop across it, the rise in cooling-air temperature through it, and the resultant density changes would have had to be taken into account.

SPECIFIED

CONCLUDING REMARKS

The pressure loss through a duct component is affected by the nature of the entering flow and, when unsymmetrical velocity distributions occur, the pressure-loss coefficients are higher than those given herein for conditions of uniform flow. This consideration raises the question of the accuracy with which the over-all losses for a duct system can be predicted by summation of component losses obtained from the material in this report. As yet no satisfactory answer to this question exists, but this lack of data in no way impairs the usefulness of the material contained herein for designing duct systems for a minimum of loss.

Although the pressure losses in a well-designed duct system should be small compared with the unavoidable heat-exchanger pressure drop, the margin of pressure available over pressure required is very small, particularly for full-power climb; and elimination of unnecessary duct losses often makes the difference between an acceptable and an unacceptable installation.

Langley Memorial Aeronautical Laboratory
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TABLE I.- ORDINATES FOR KRÖBER VANE PROFILES

x/c	y/c			
	90° bend	60° bend	45° bend	30° bend
0.00	0.000	0.000	0.000	0.000
.05	.087	.041	-----	-----
.10	.154	.074	.044	.031
.15	.200	.100	-----	-----
.20	.236	.124	.075	.051
.25	.262	.140	-----	-----
.30	.277	.153	.094	.067
.35	.284	.161	-----	-----
.40	.284	.166	.105	.071
.45	.283	.168	-----	-----
.50	.273	.164	.103	.071
.55	.260	.157	-----	-----
.60	.242	.151	.094	.067
.65	.219	.142	-----	-----
.70	.192	.129	.078	.055
.75	.167	.111	-----	-----
.80	.137	.096	.058	.043
.85	.104	.072	-----	-----
.90	.071	.048	.030	.024
.95	.037	.026	-----	-----
1.00	.000	.000	.000	.000

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TABLE II.- ESTIMATION OF TOTAL-PRESSURE-LOSS COEFFICIENTS FOR DUCT SYSTEM I

[Mass flow = 0.109 slug/sec; temperature = 584.4° F abs.]

Initial station	Final station	Controlling parameters			Calculated values	
Duct component, rectangular diffusers						
		Diffuser equivalent conical angle of expansion, 2θ (deg)	Initial-station cross-sectional area, A_{di} (sq ft)	Final-station cross-sectional area, A_{de} (sq ft)	Diffuser coefficient, k_2 (fig. 8)	Diffuser total-pressure-loss coefficient, $\Delta H/q$ (1)
1	2	7.0	0.250	0.515	0.130	0.034
3	4	13.8	.515	3.000	.267	.183
Duct component, 90° rectangular bends						
		Bend aspect ratio, h/w	Bend radius ratio, $\bar{r}/w$	$\frac{\text{Mass flow}}{\text{Perimeter}}$, m/P $\left(\frac{\text{slug/sec}}{\text{ft}}\right)$	Reynolds number, R (fig. 1(b))	Bend total-pressure-loss coefficient, $\Delta H/q$ (fig. 2)
2	3	1.0	3.00	0.0330	370,000	0.069
4	5	1.0	.78	.0158	155,000	.500

[1]Diffuser total-pressure-loss coefficient $\frac{\Delta H}{q} = k_2\left(1 - \frac{A_{di}}{A_{de}}\right)^2$.

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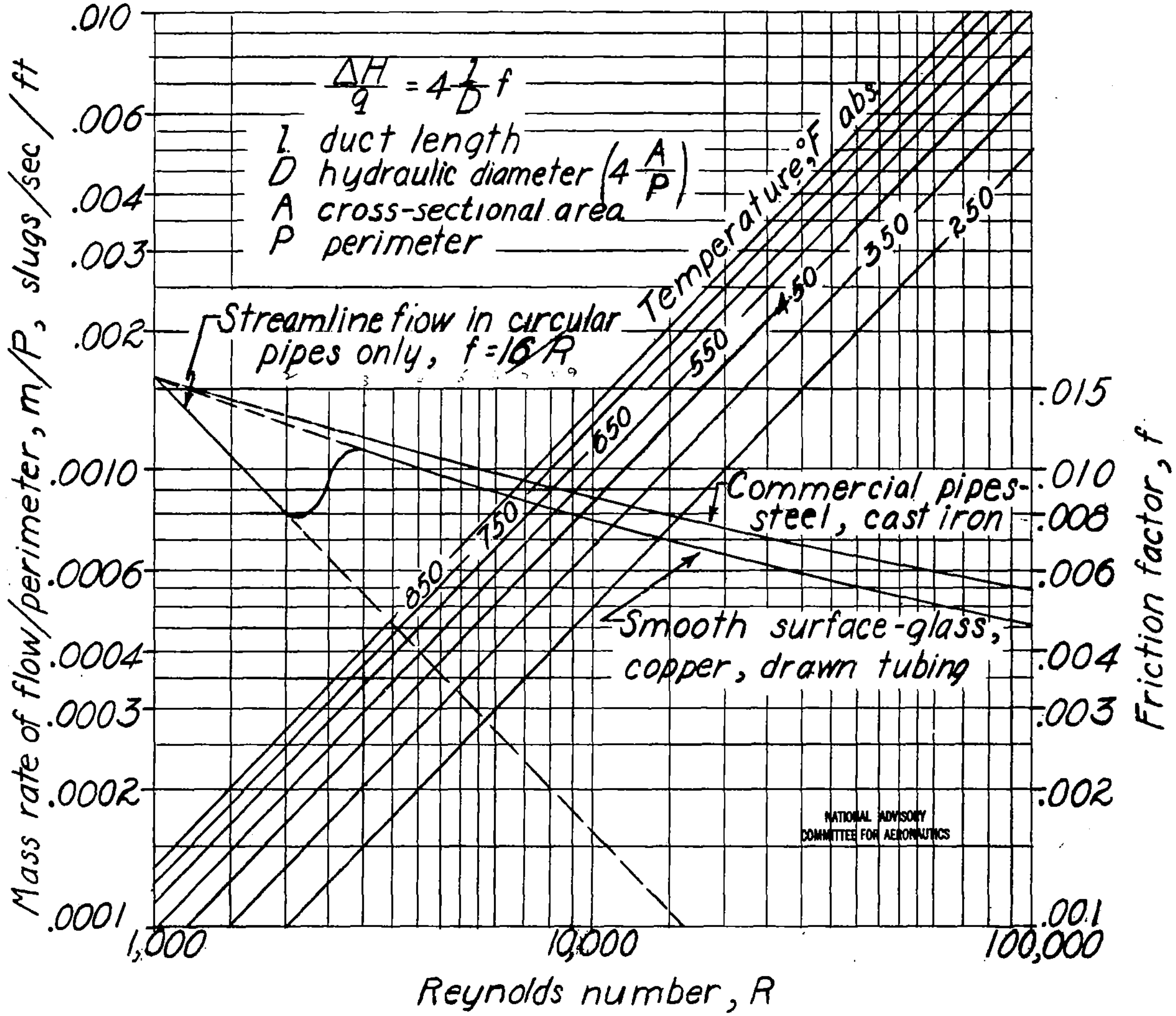
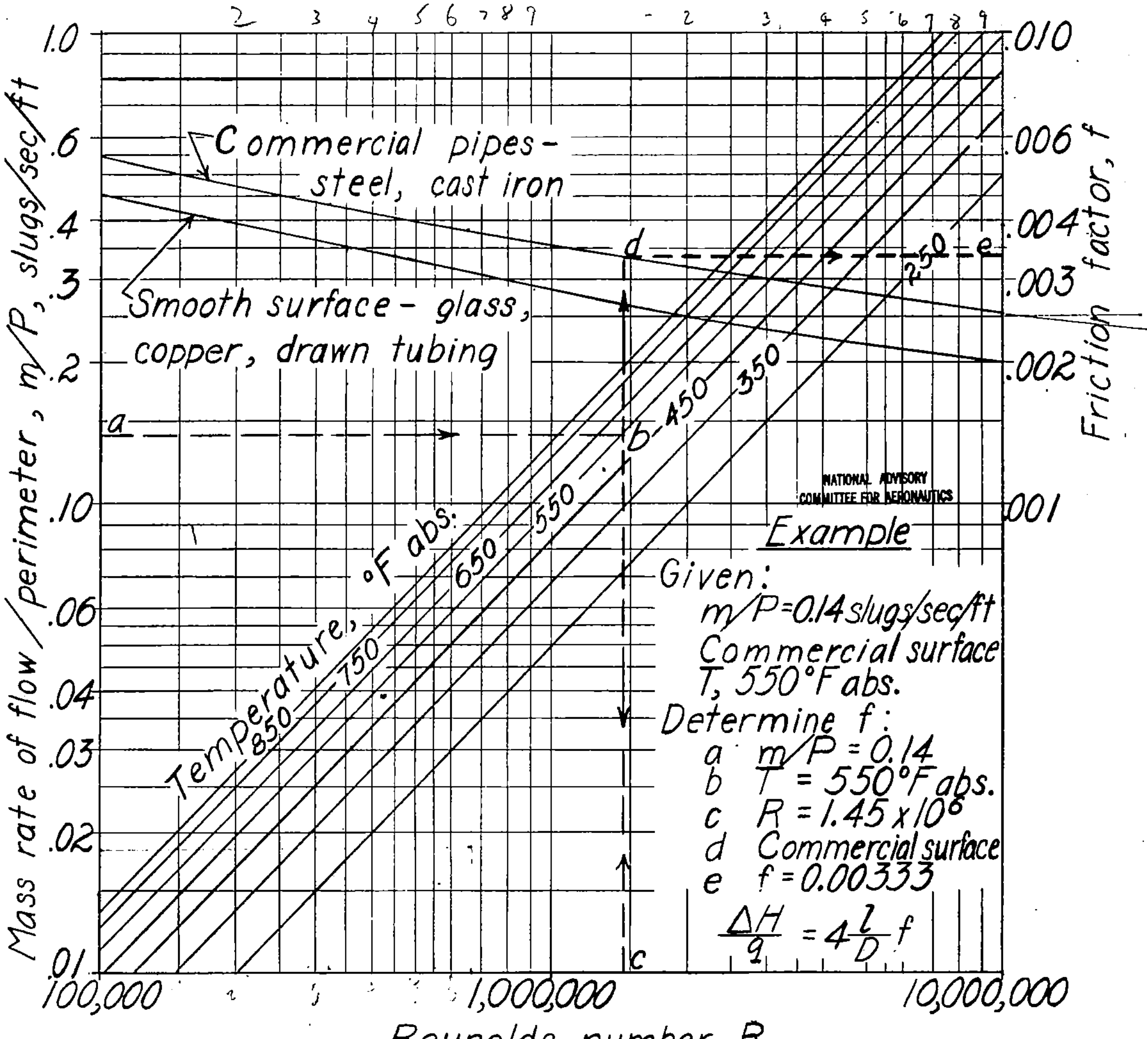


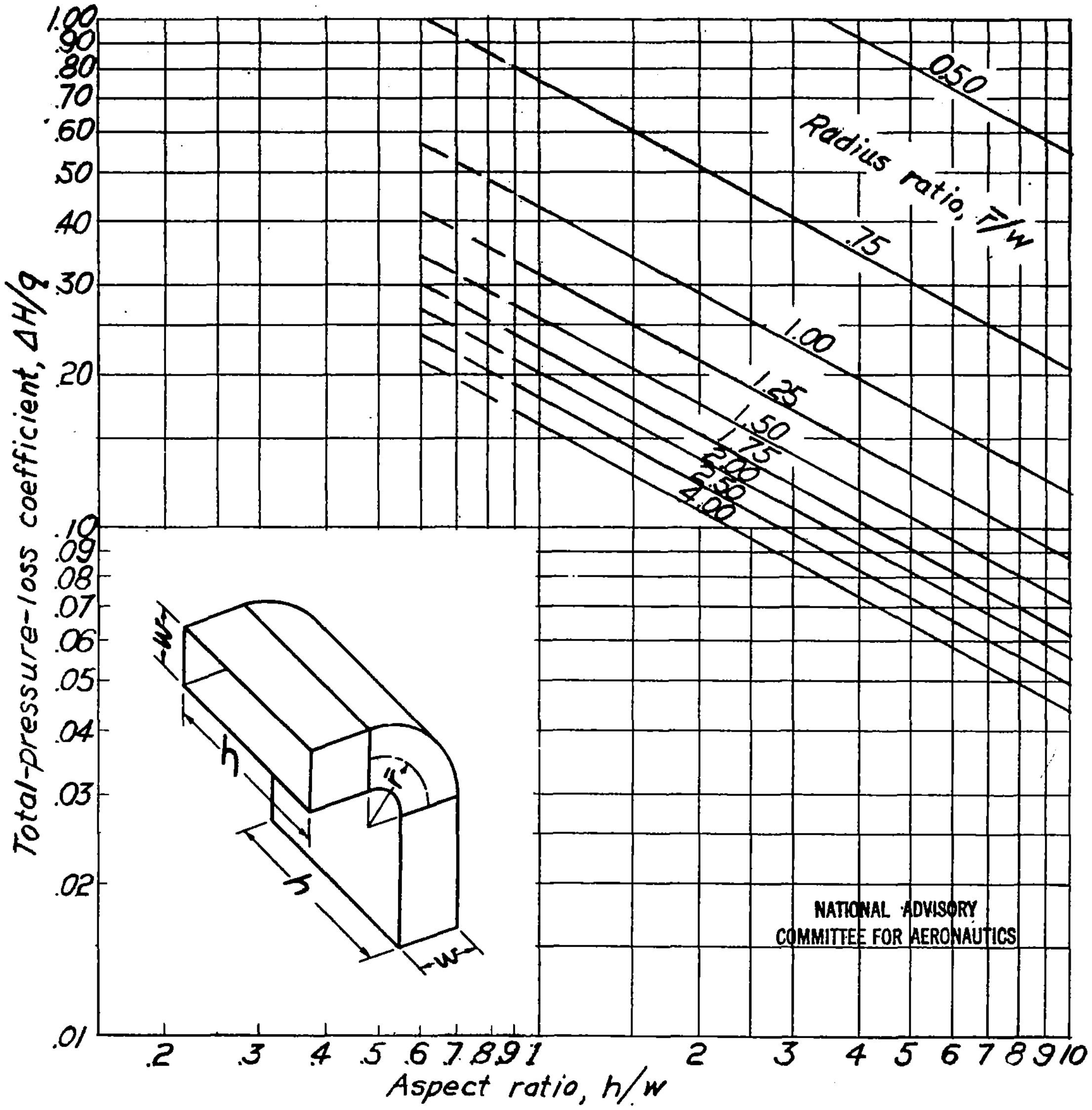
(a) Reynolds number, 1,000 to 100,000.

Figure 1.-Friction-factor and Reynolds number determination for straight ducts.

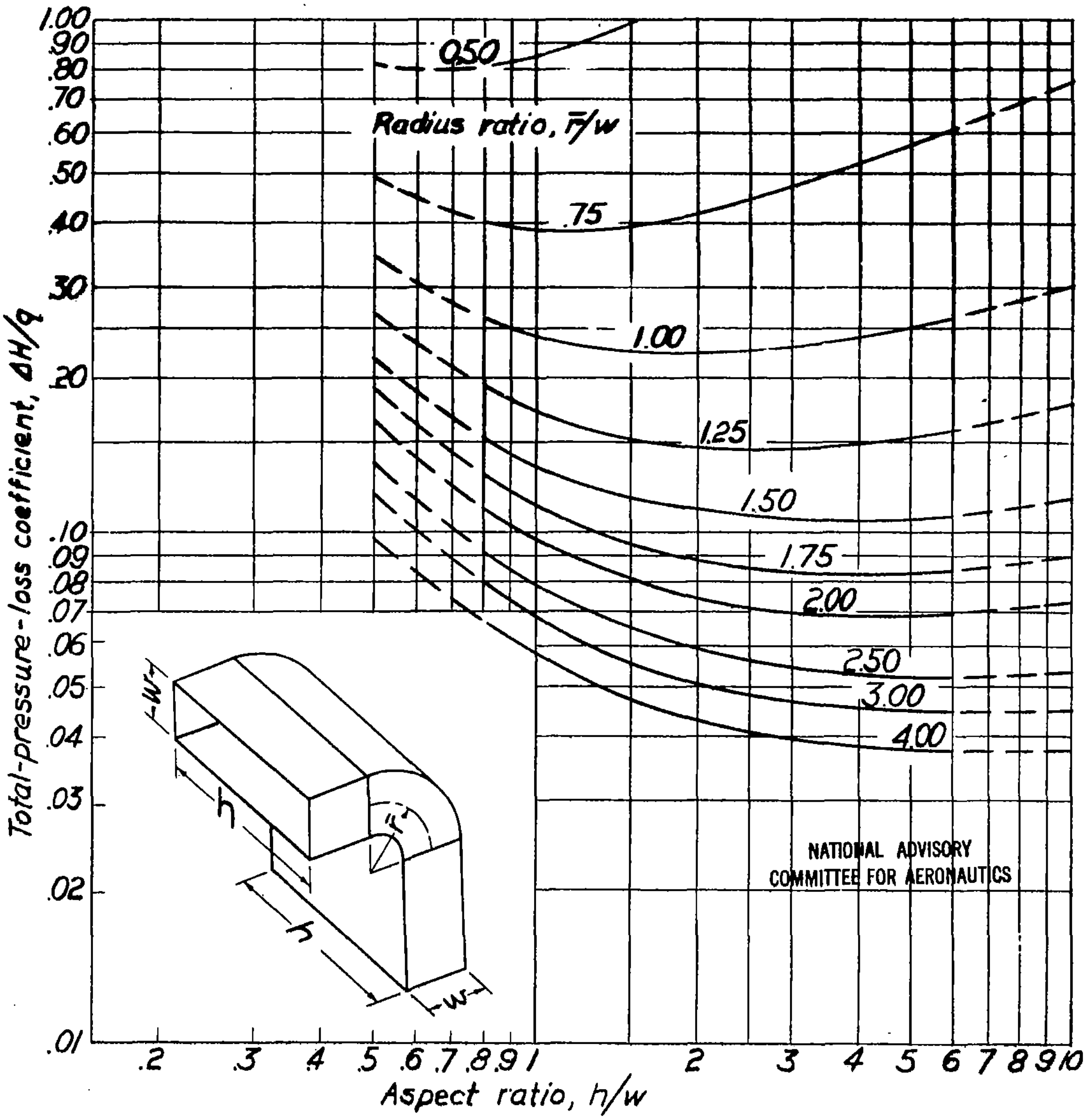


(b) Reynolds number, 100,000 to 10,000,000.

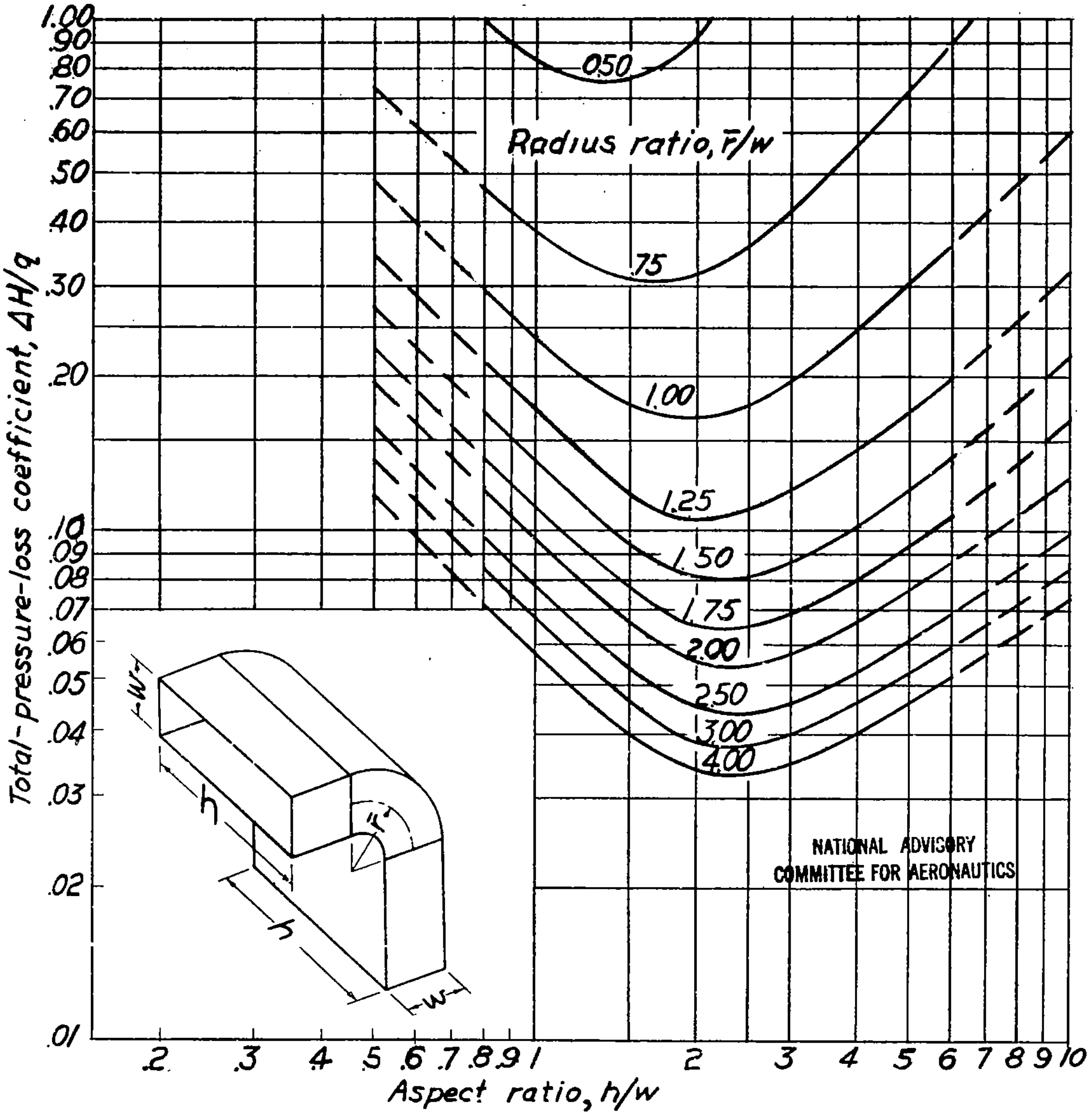
Figure 1.- Concluded.



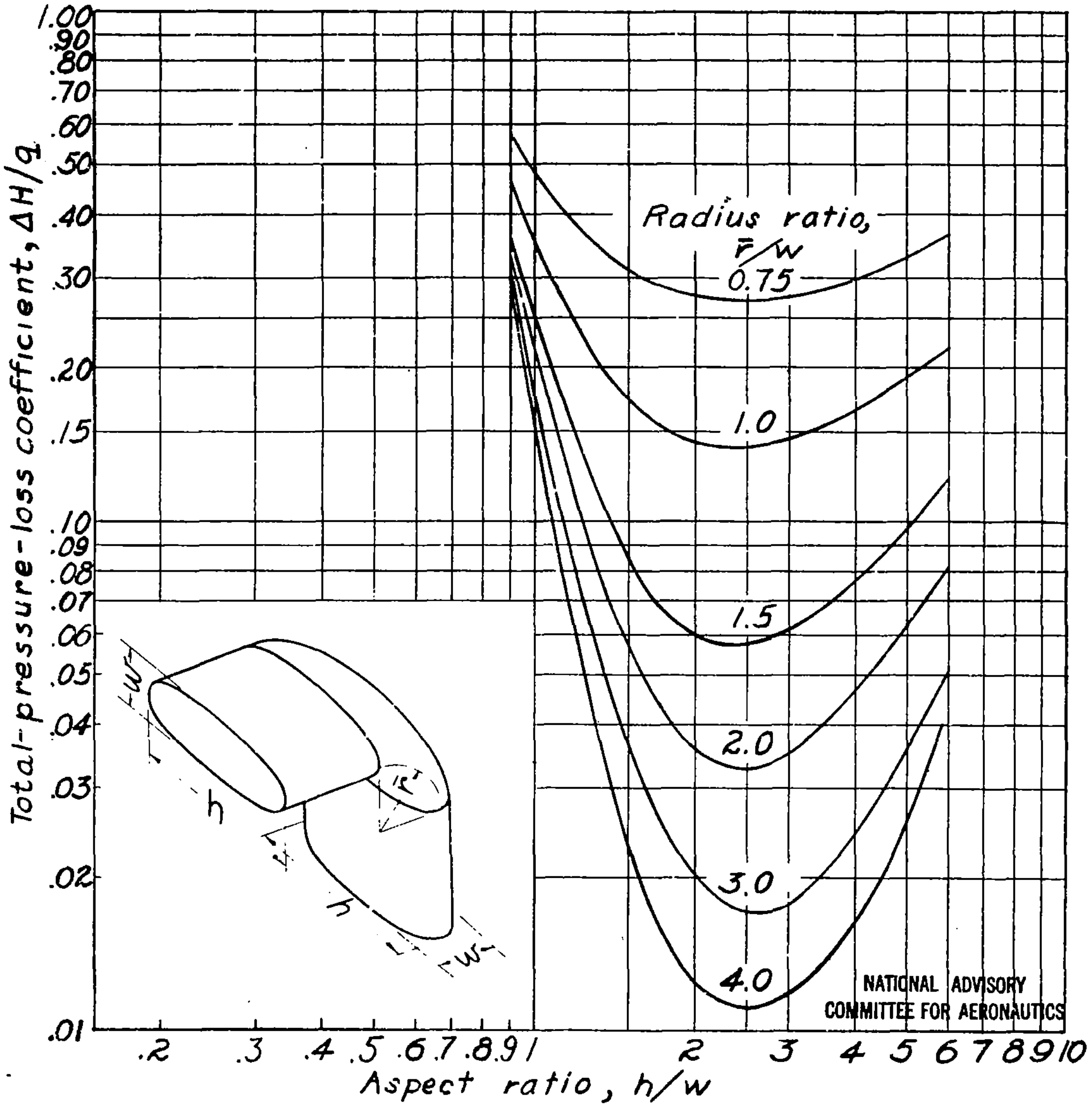
(a) Reynolds number, 100,000.
Figure 2.—Total-pressure-loss coefficients for rectangular 90° bends.



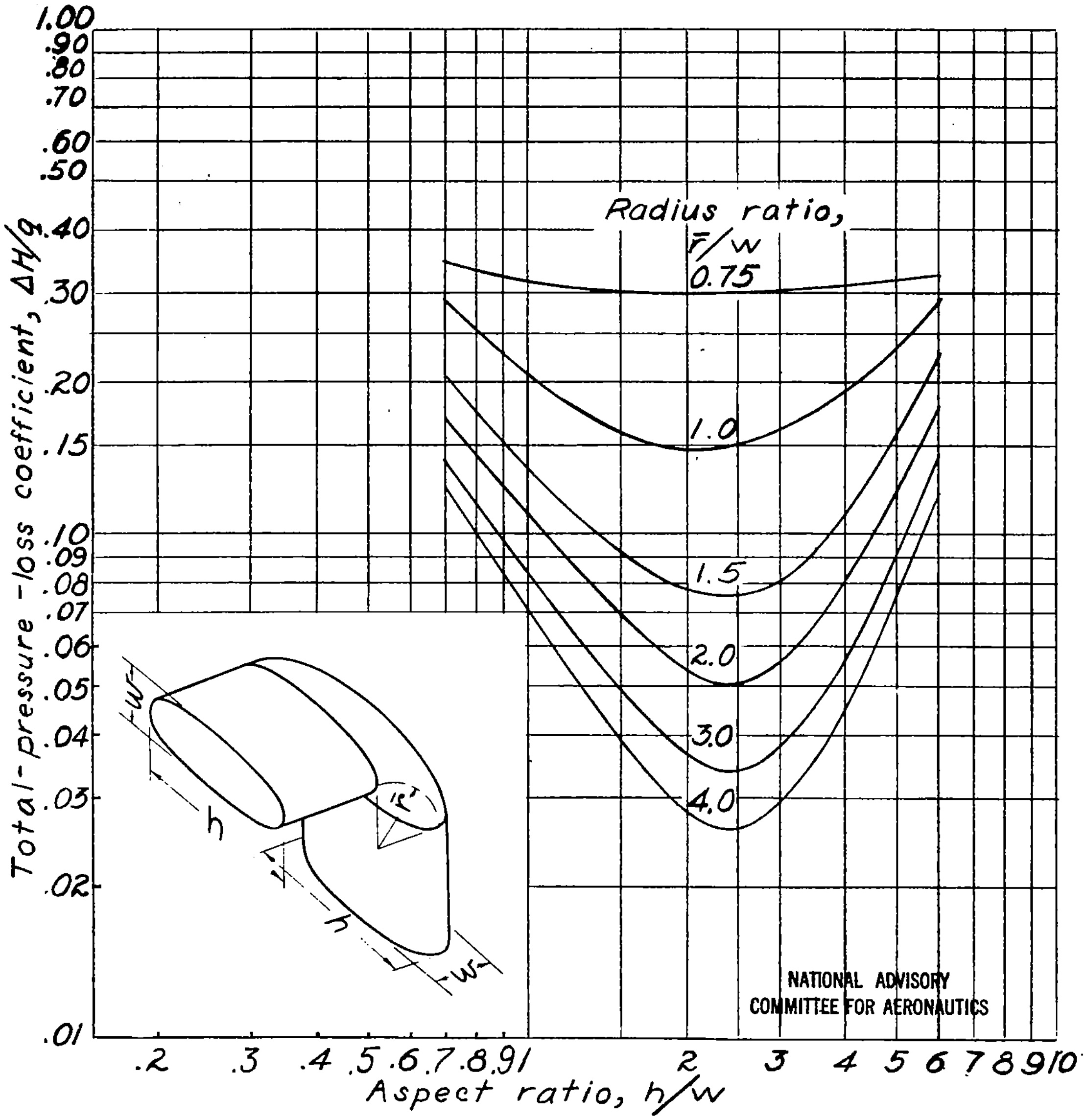
(b) Reynolds number, 300,000.
Figure 2.- Continued.



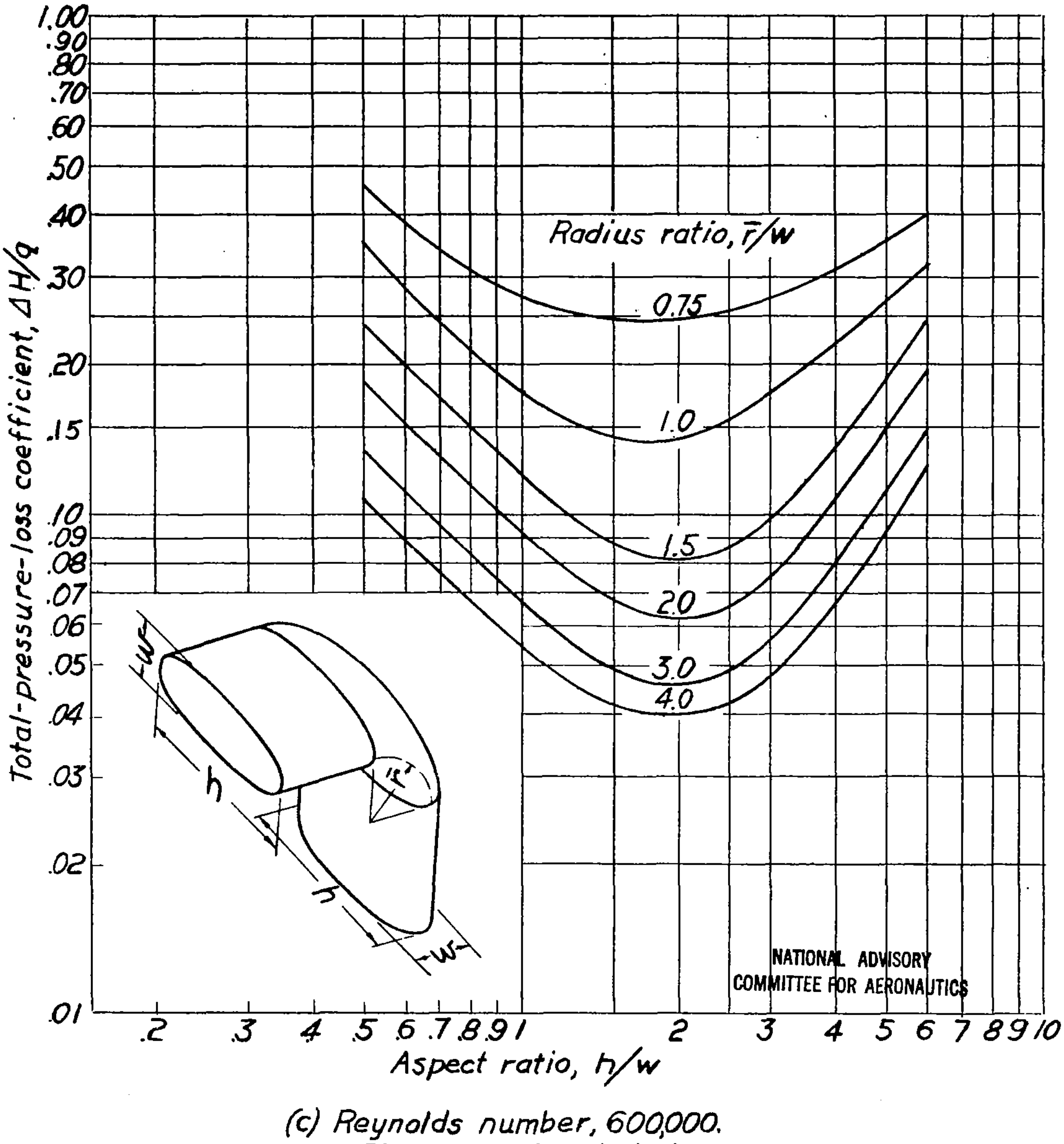
(c) Reynolds number, 600,000.
Figure 2.– Concluded.



(a) Reynolds number, 150,000.
Figure 3.- Total-pressure-loss coefficients for elliptical 90° bend.



(b) Reynolds number, 300,000.
Figure 3 .- Continued.



(c) Reynolds number, 600,000.
Figure 3.- Concluded.

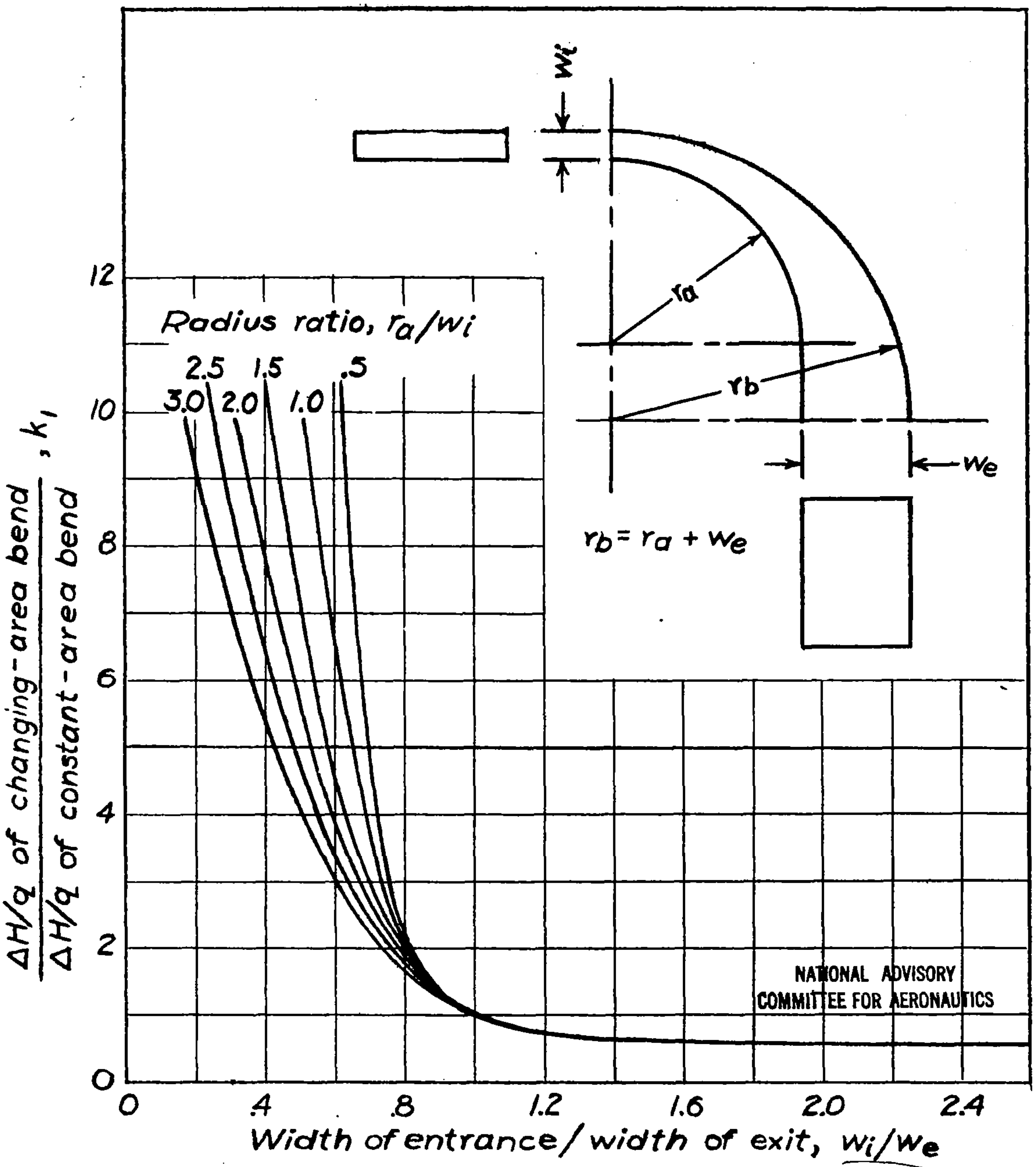


Figure 4.- Total-pressure-loss coefficient factor k_1 for 90° bend of changing area.

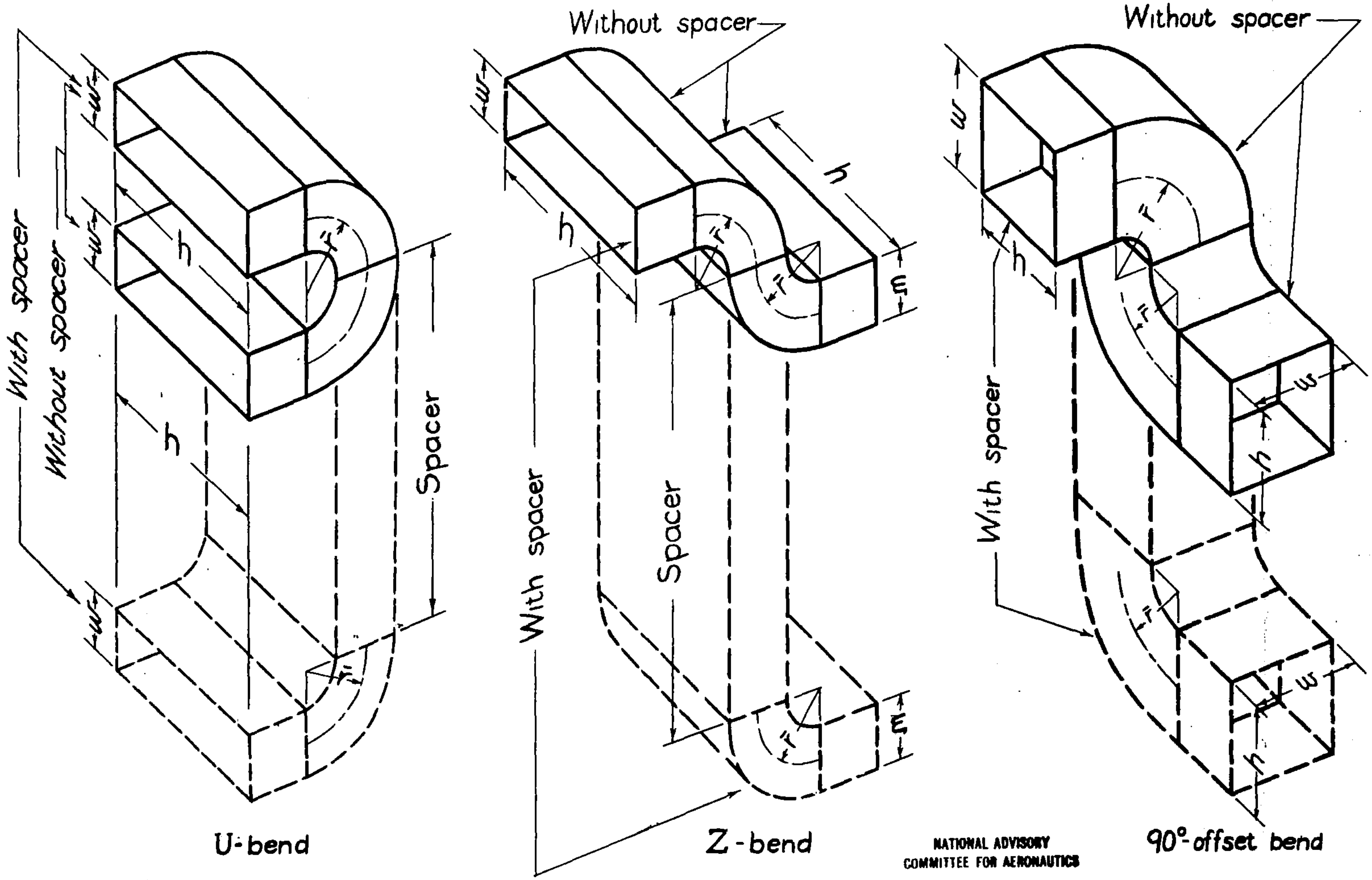
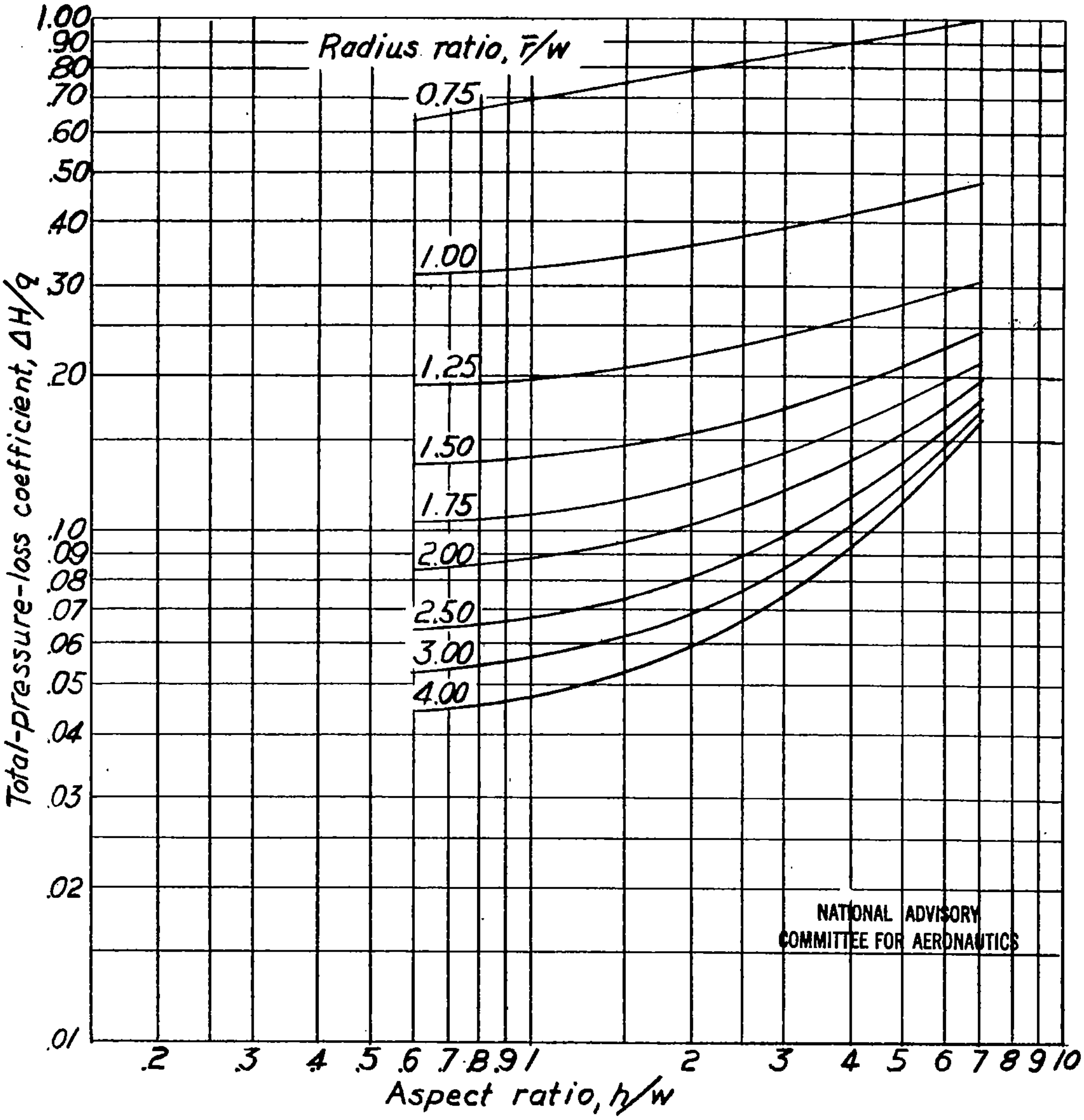
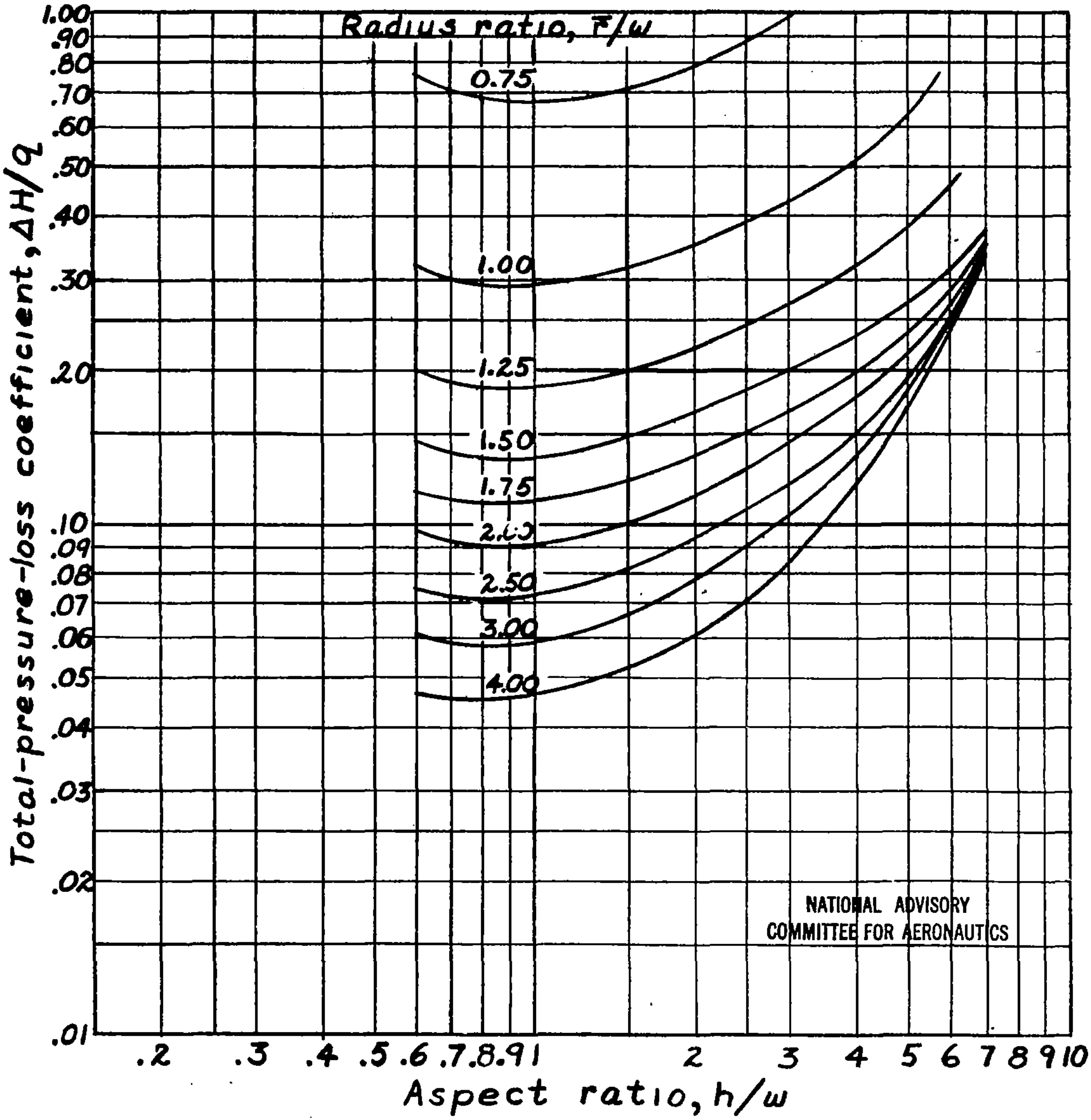


Figure 5.- Sketches of rectangular compound bends.

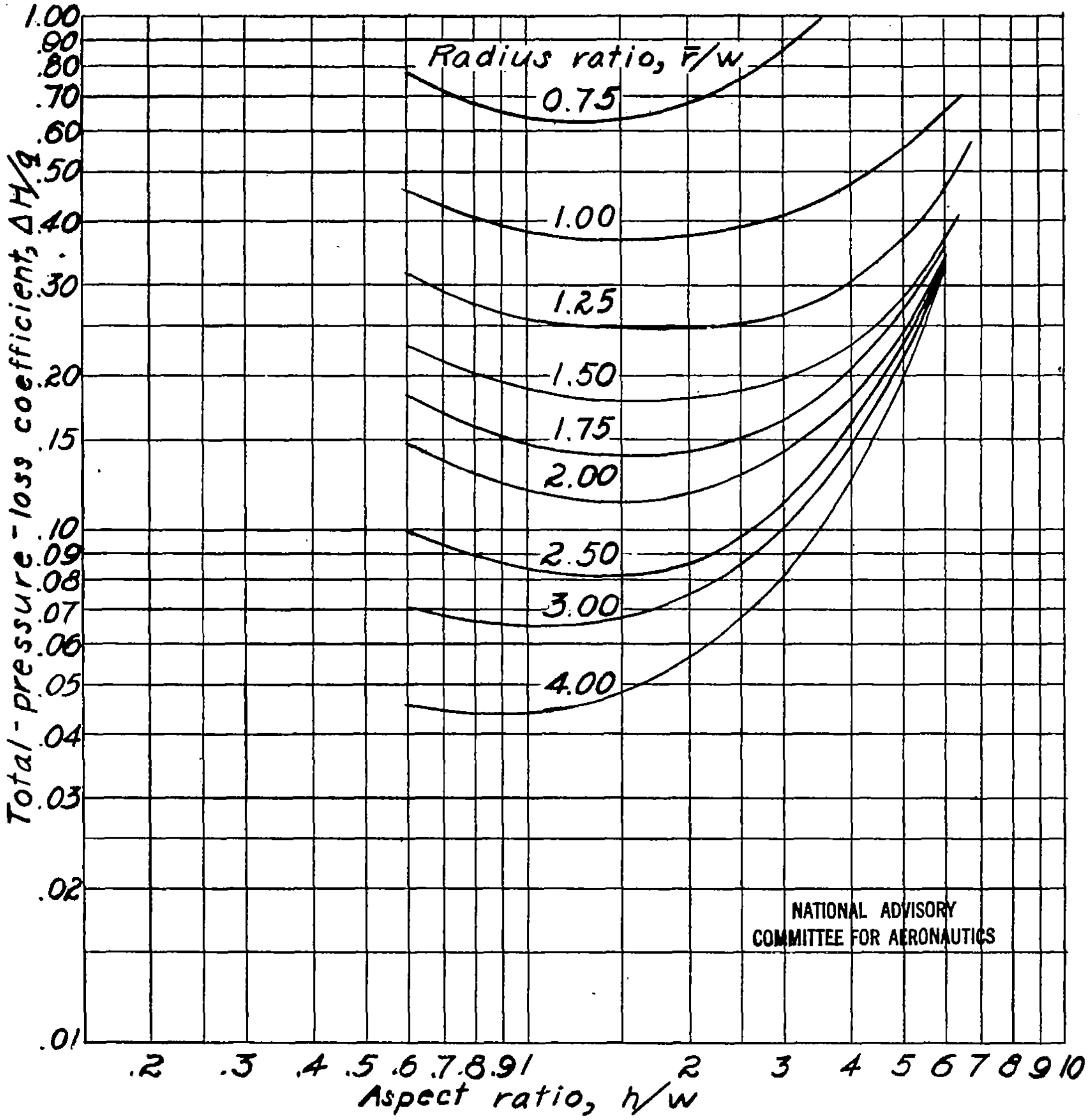


(a) Bends without spacers; Reynolds number, 300,000.
Figure 6.- Total-pressure-loss coefficients for compound rectangular U-, Z-, and 90°- offset bends.



(b) Bends without spacers; Reynolds number, 600,000.

Figure 6. - Continued.



(c) Bends with 5-foot spacer; Reynolds number, 600,000.
Figure 6.- Concluded.

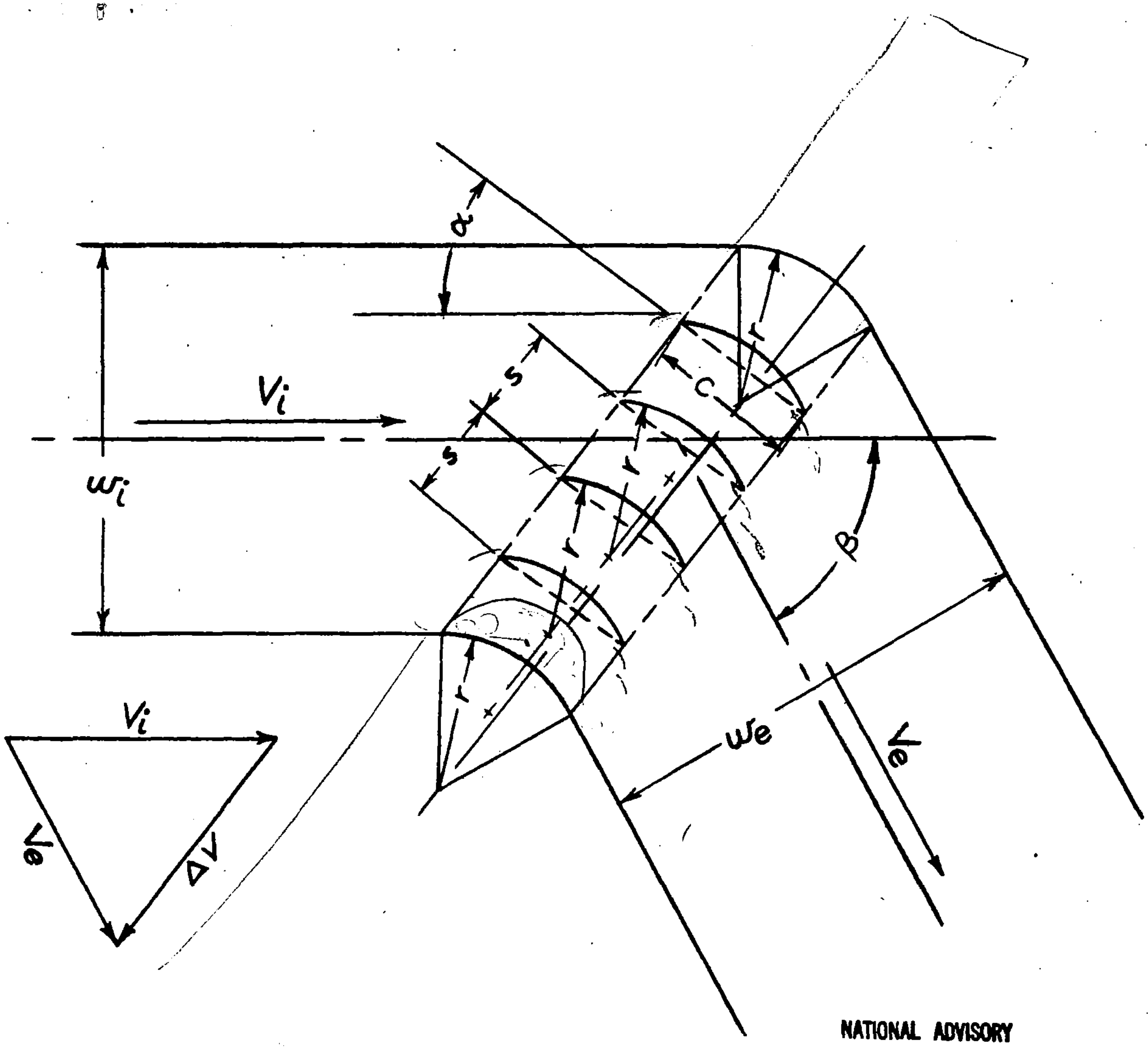
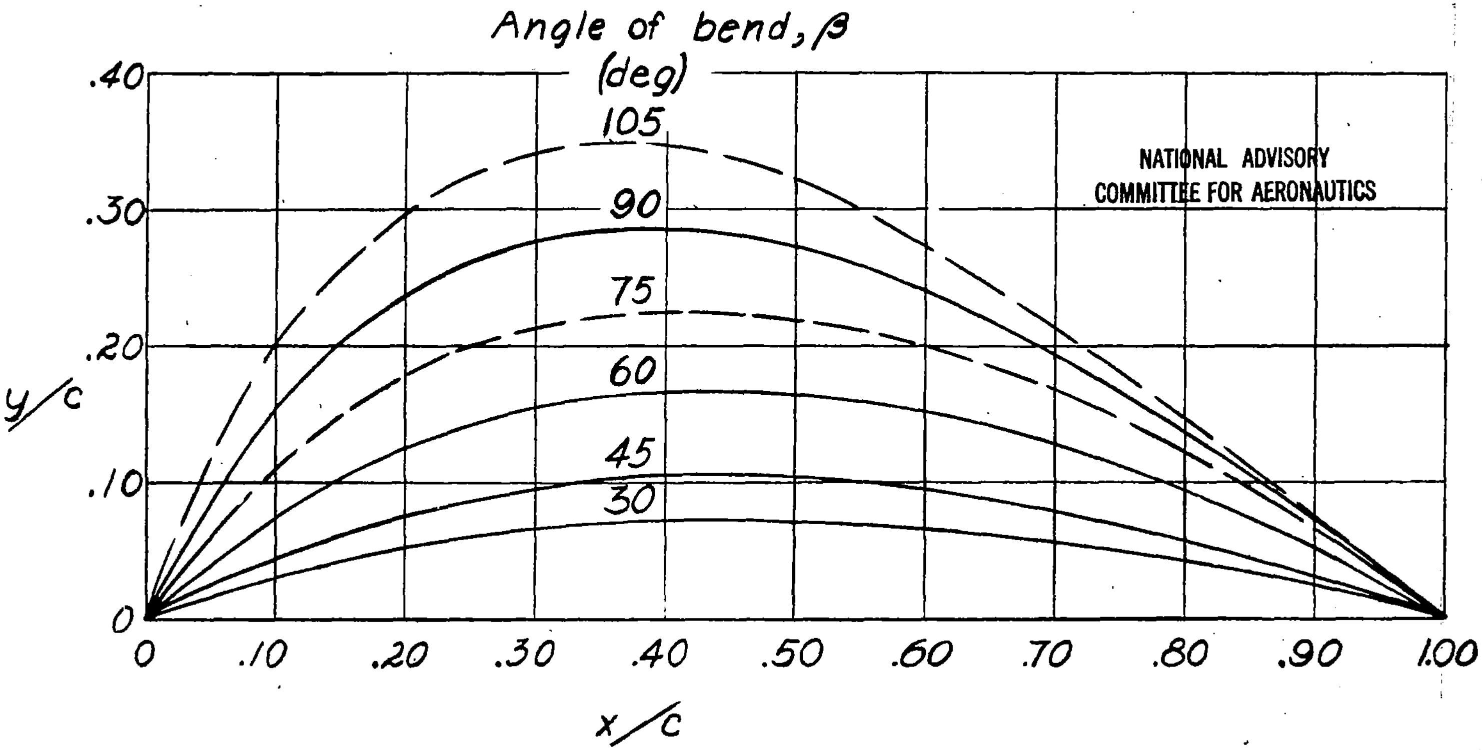
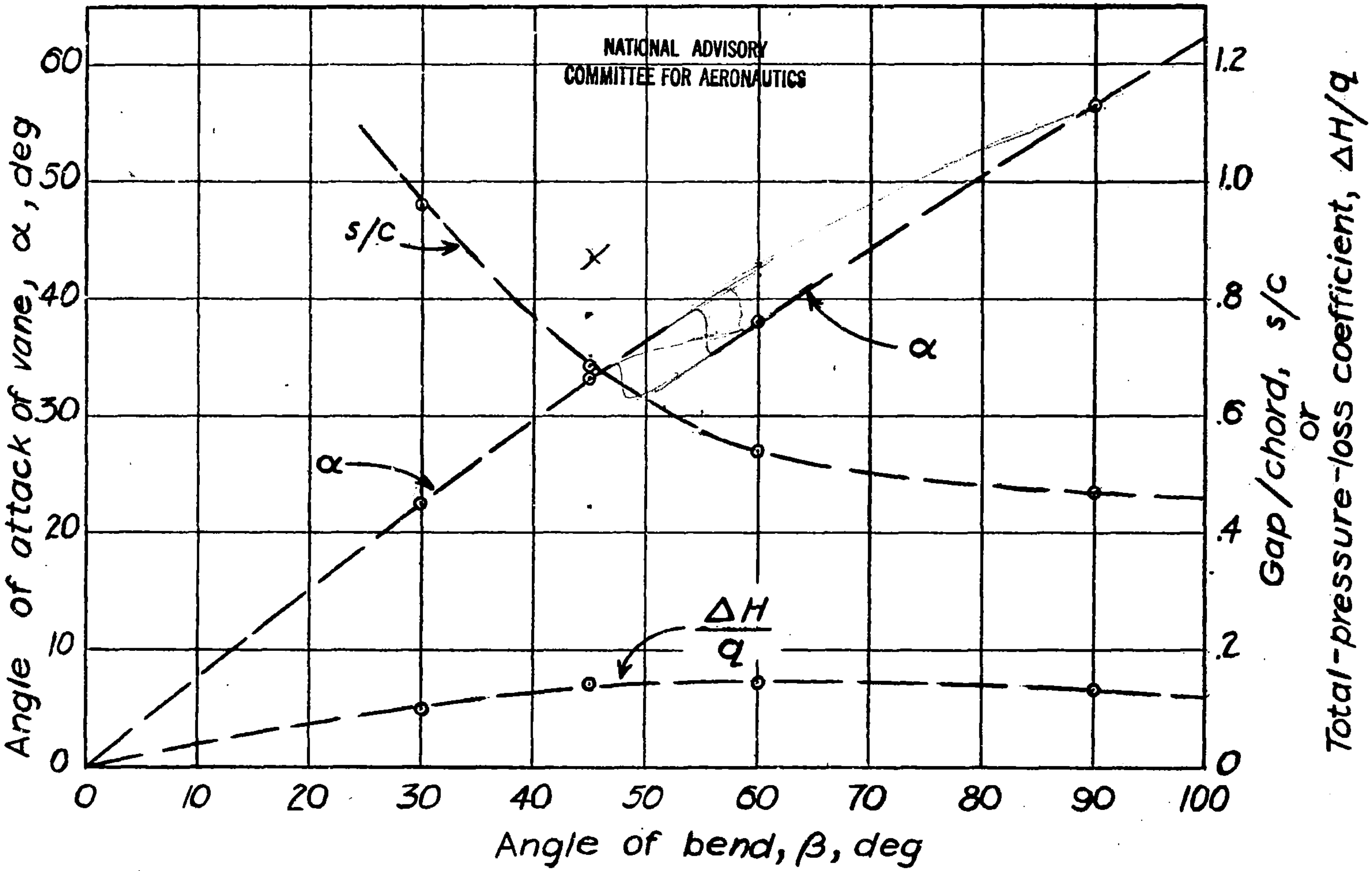


Figure 7. - Bend with thin circular-arc vanes.



(a) Vane profiles (x and y, coordinates of points on vane profile).

Figure 8.- Design data for Kröber thin vanes. (Data for β = 30°, 45°, 60°, and 90° taken from reference 9.)



(b) Vane characteristics at a Reynolds number of 40,000.

Figure 8 .- Concluded.

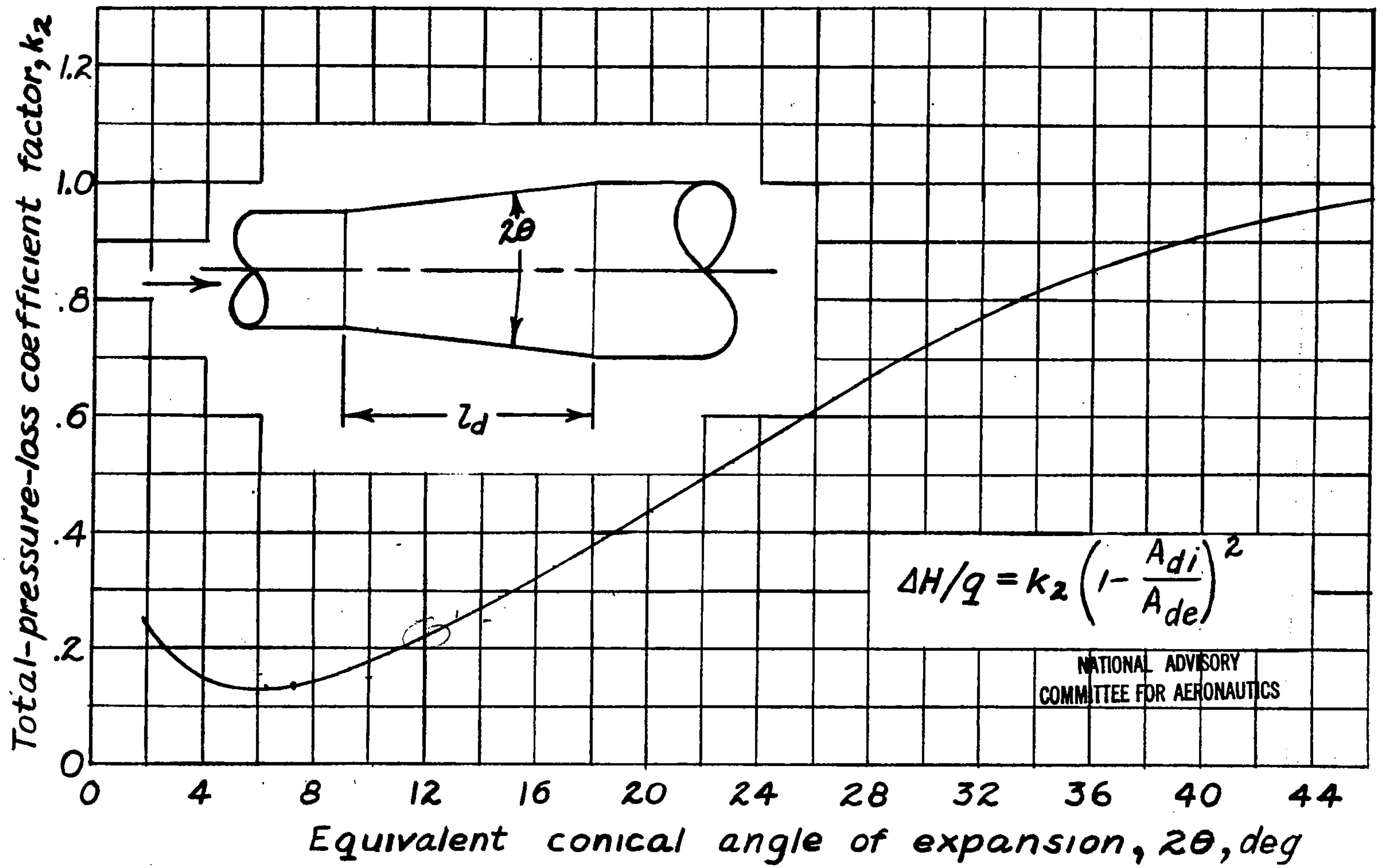


Figure 9.– Total-pressure-loss coefficient factor k_2 for straight-wall conical diffusers.

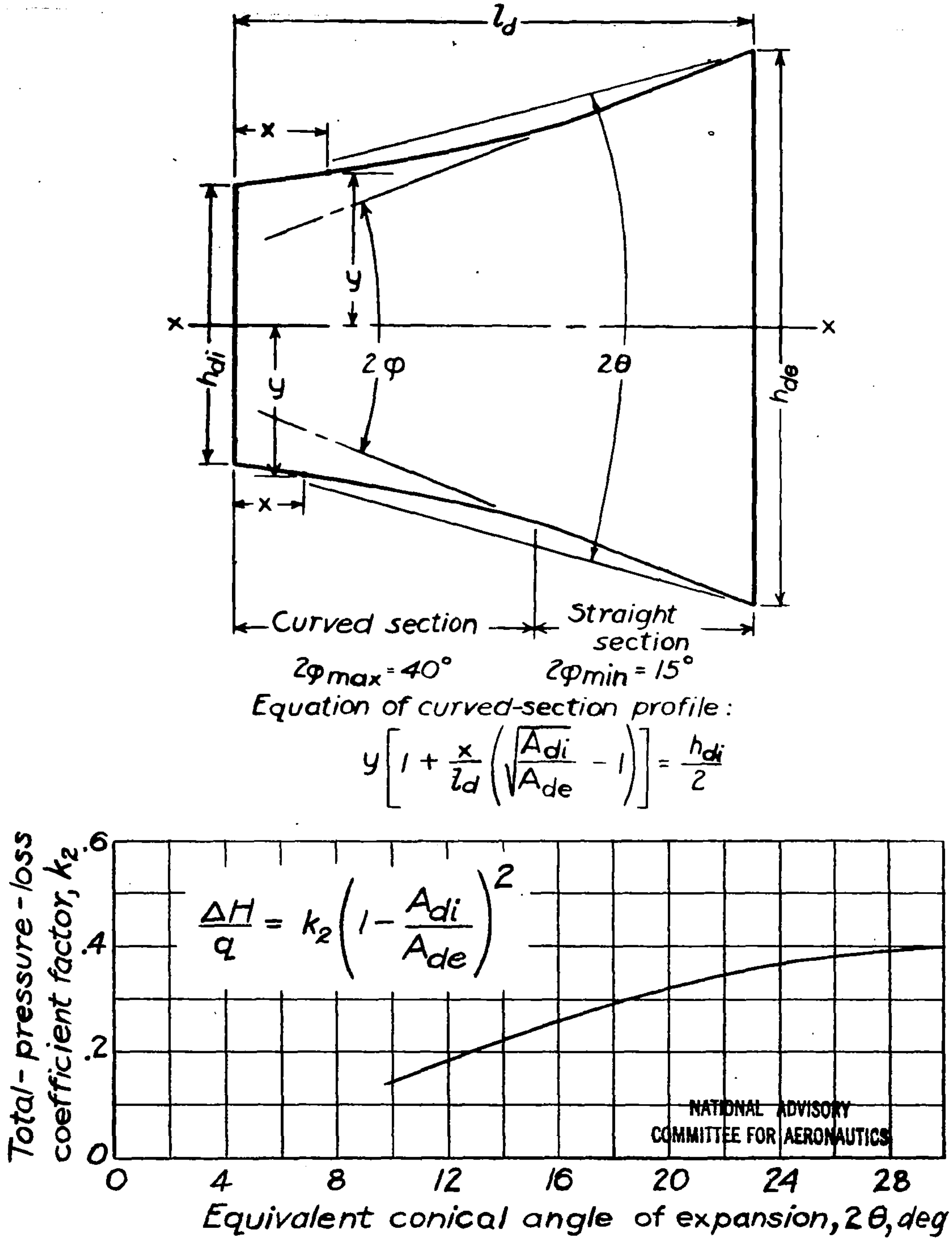


Figure 10.- Total-pressure-loss coefficient factor k_2 for curved-wall conical diffusers.

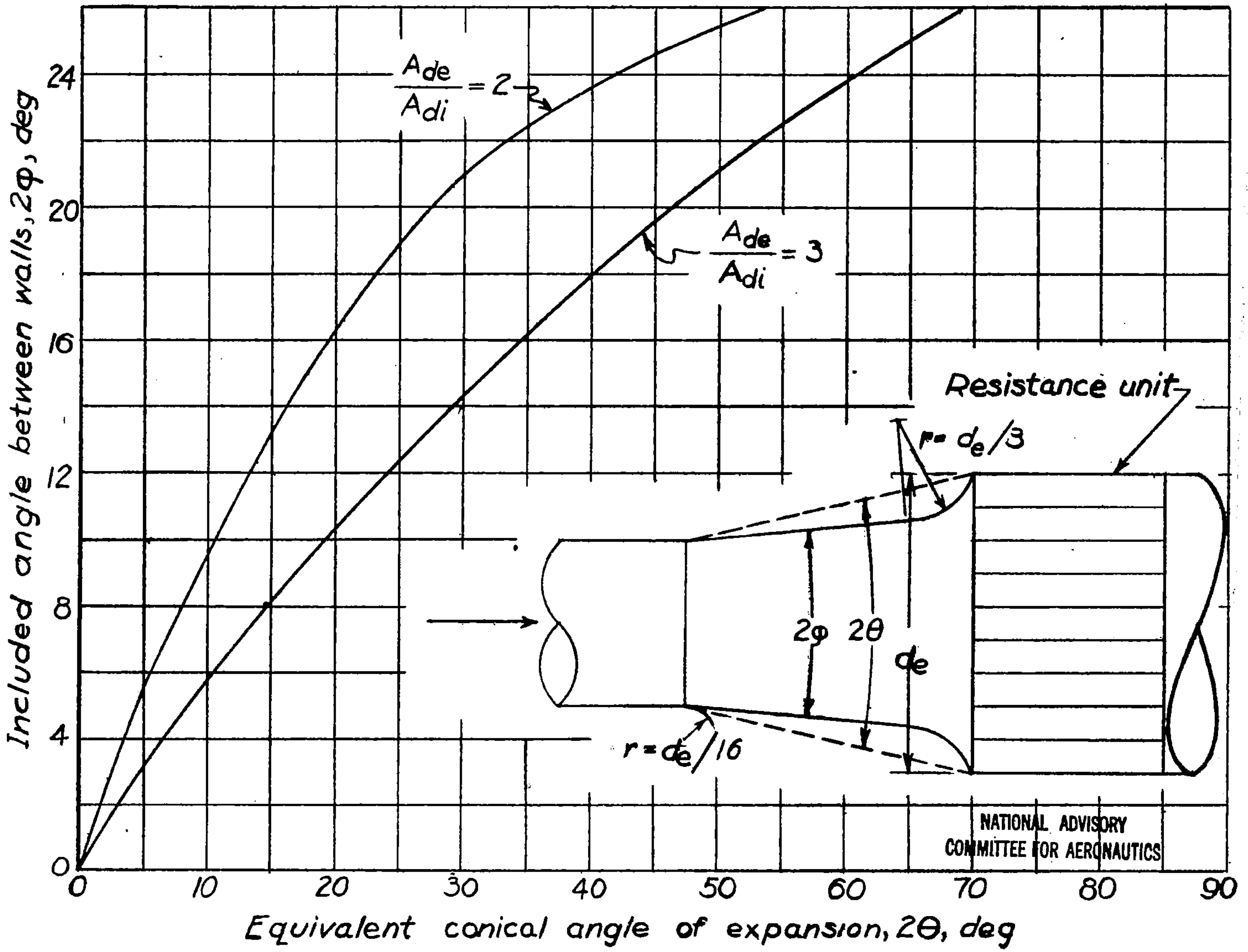


Figure 11.- Design of conical diffusers followed by resistance units.

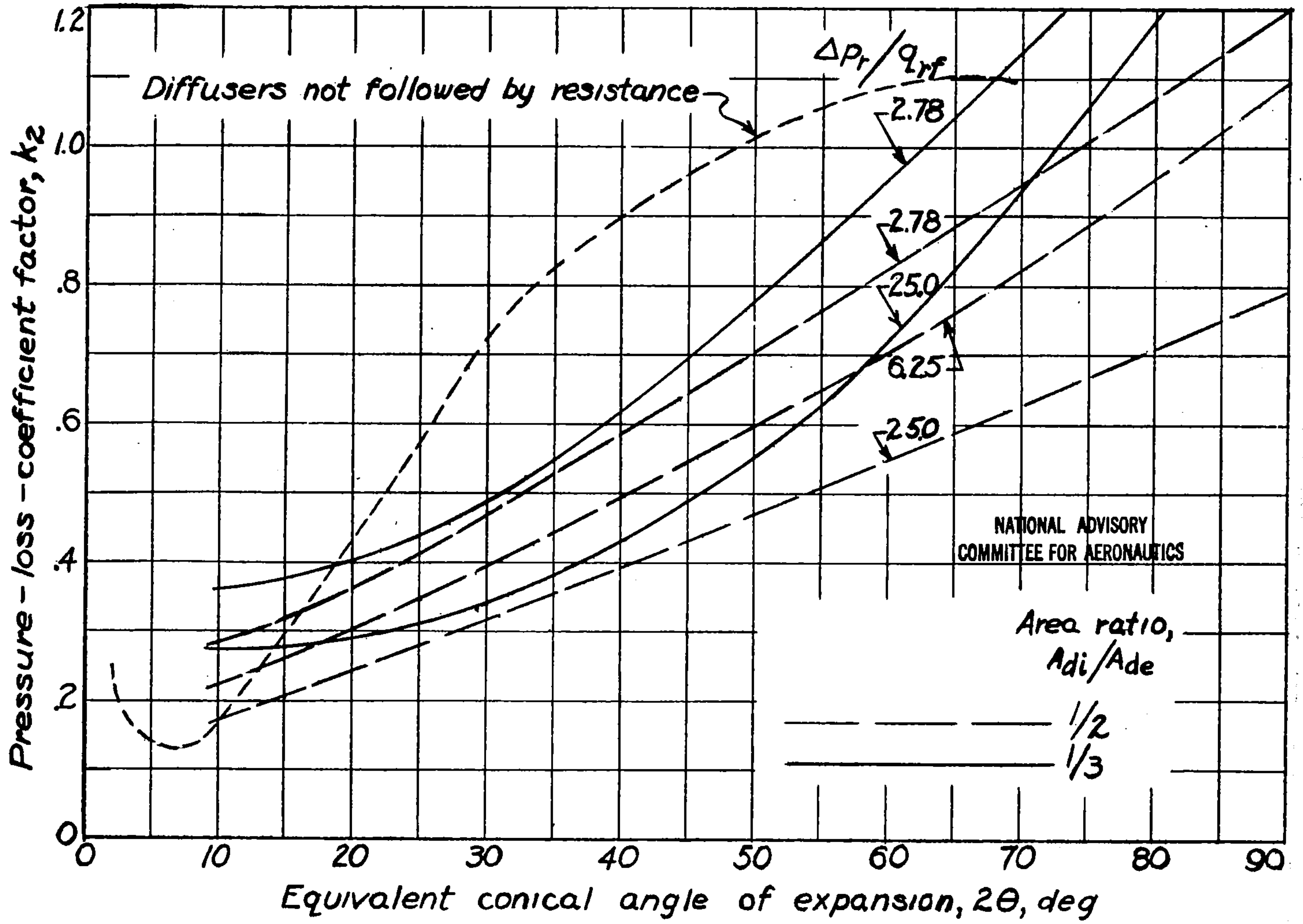
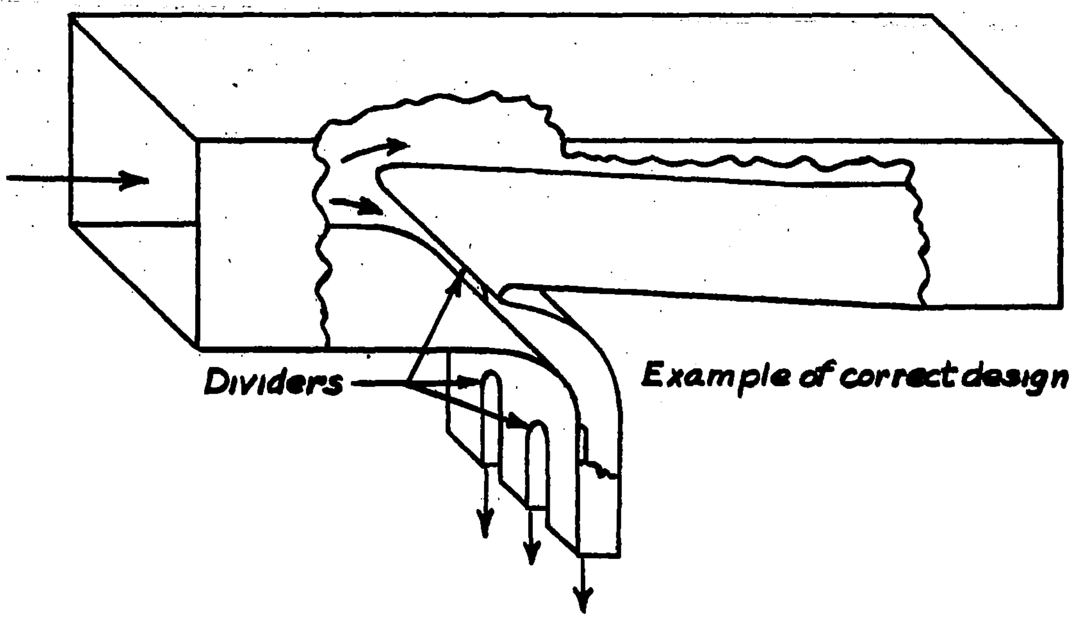
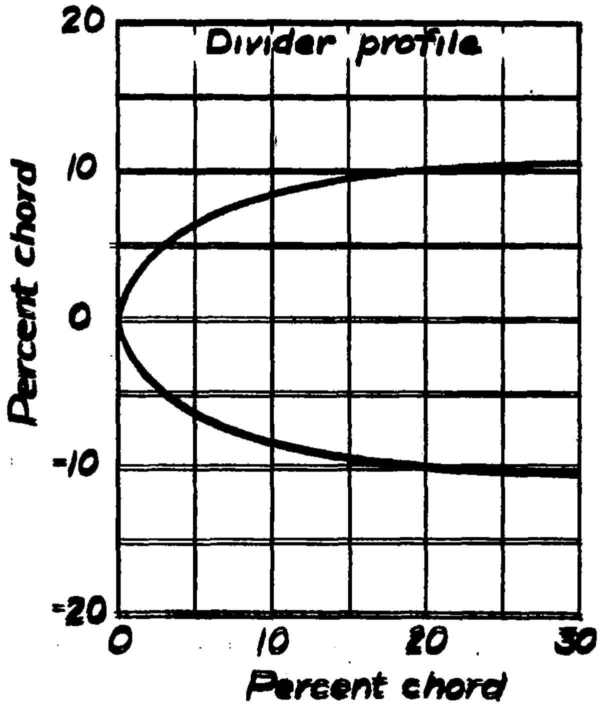


Figure 12.-Total-pressure-loss coefficient factor k_2 for conical diffusers illustrated in figure 11.





Divider-profile ordinates		
Station (Percent chord)	Upper surface	Lower surface
0	0	0
1.25	3.31	-3.31
2.50	4.58	-4.58
5.00	6.22	-6.22
7.50	7.35	-7.35
10.00	8.20	-8.20
15.00	9.35	-9.35
20.00	10.04	-10.04
25.00	10.40	-10.40
30.00	10.50	-10.50

Figure 13.- Branch-duct design (divider profile, NACA 0021 airfoil nose section).

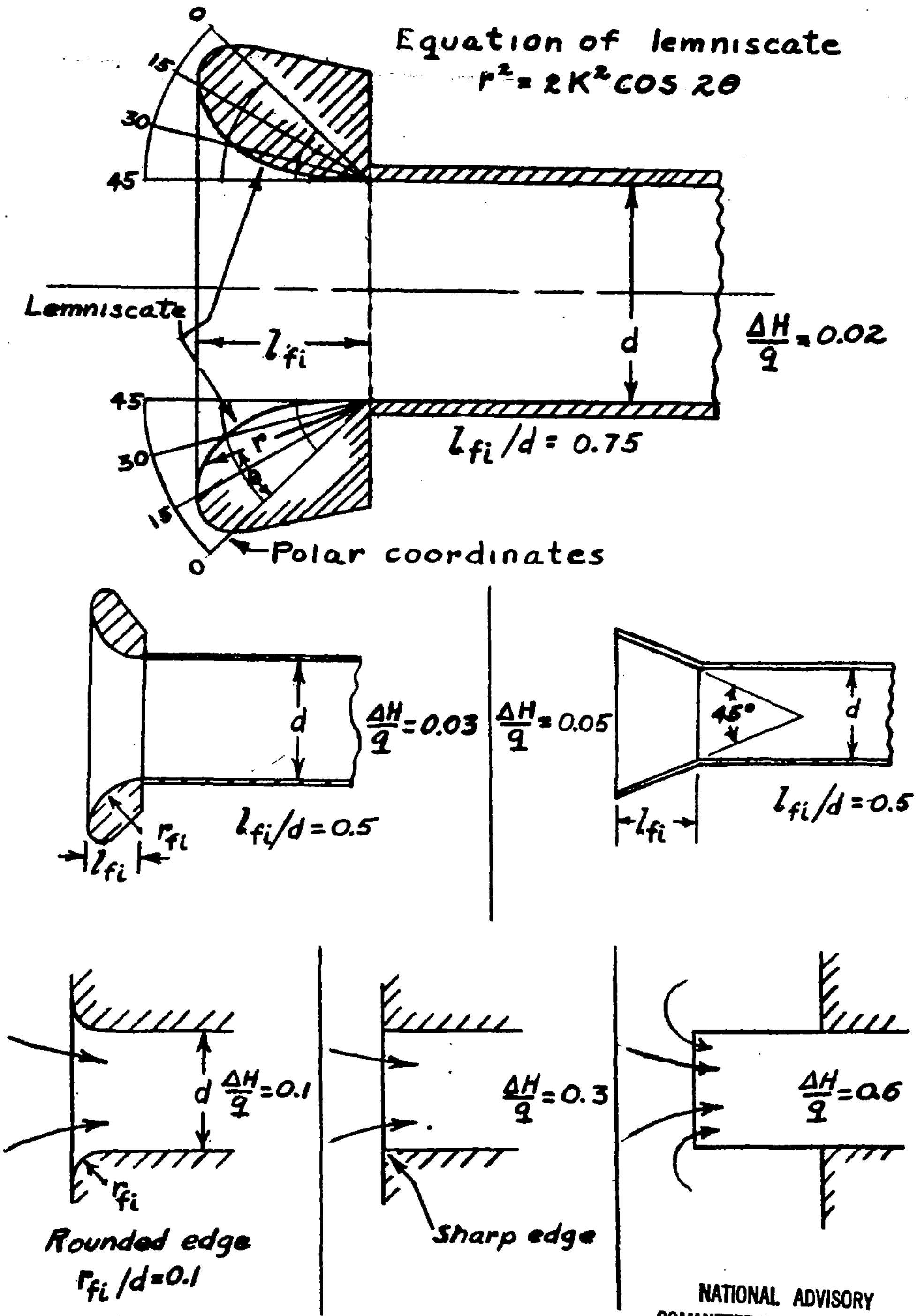


Figure 14.- Internal-duct-inlet designs and total-pressure-loss coefficients.

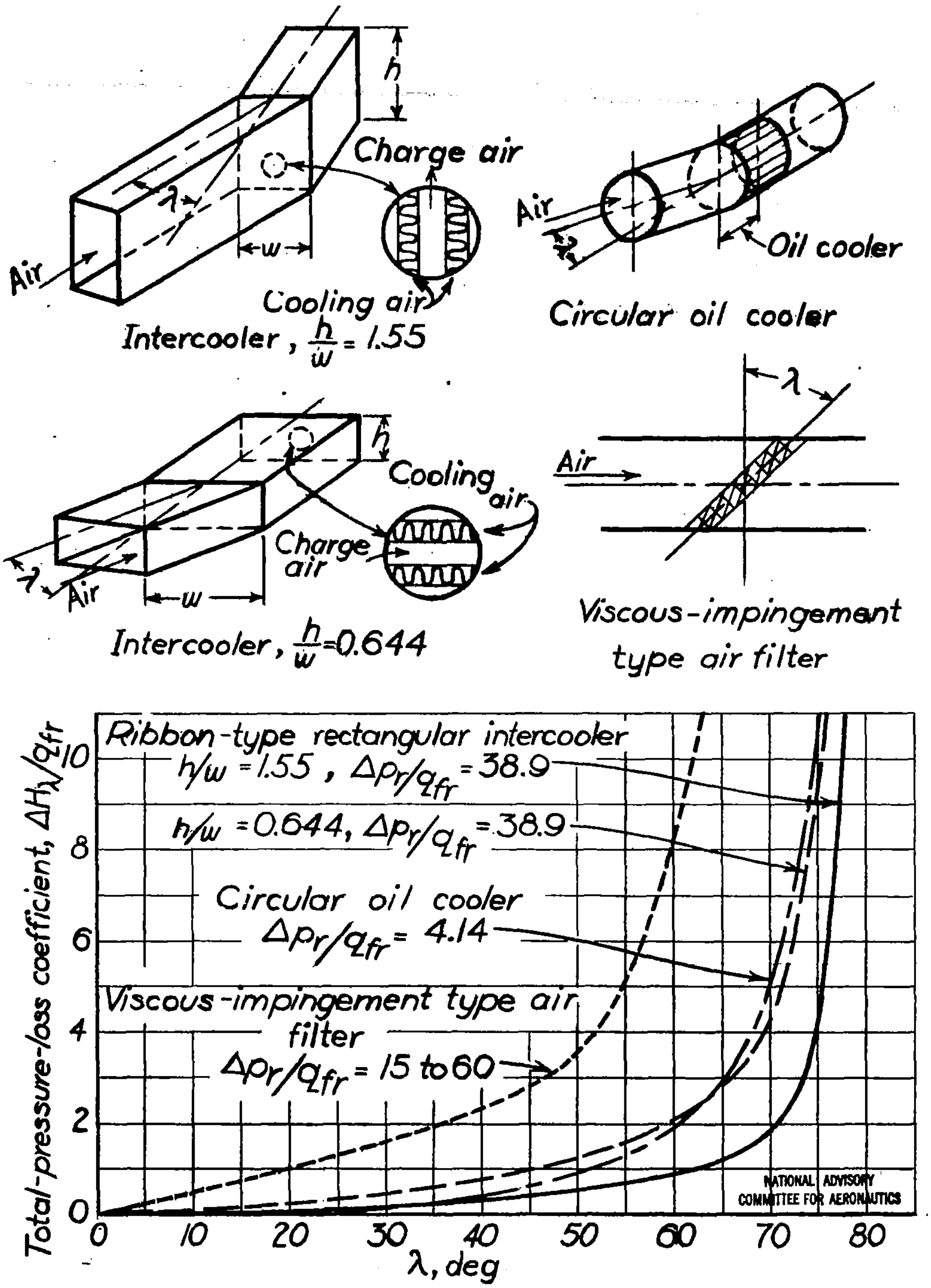


Figure 15.- Total-pressure-loss coefficients for resistance units set at an angle to the upstream duct (average V_{fr}, 16.5 feet per second).

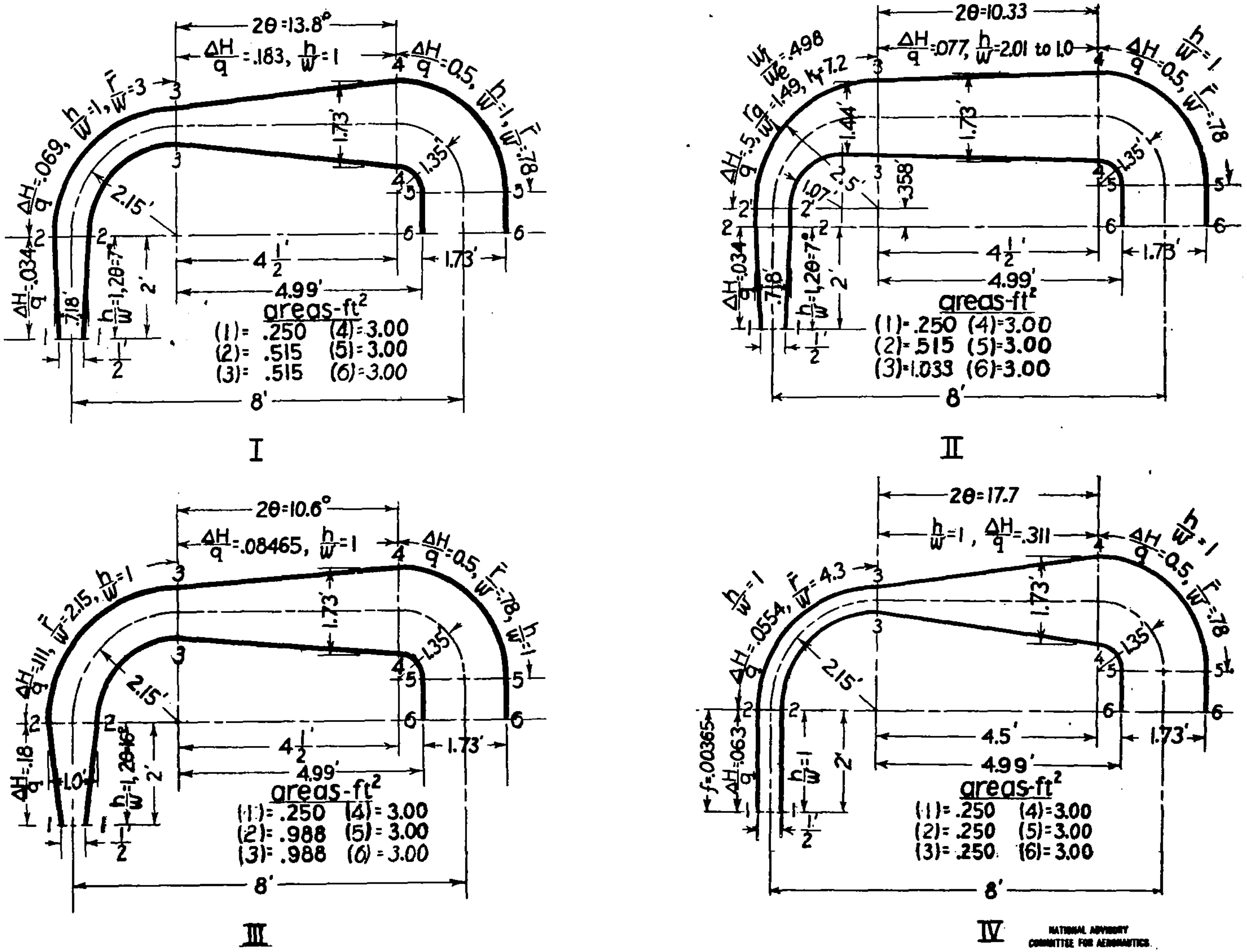


Figure 16.- Sample duct systems.

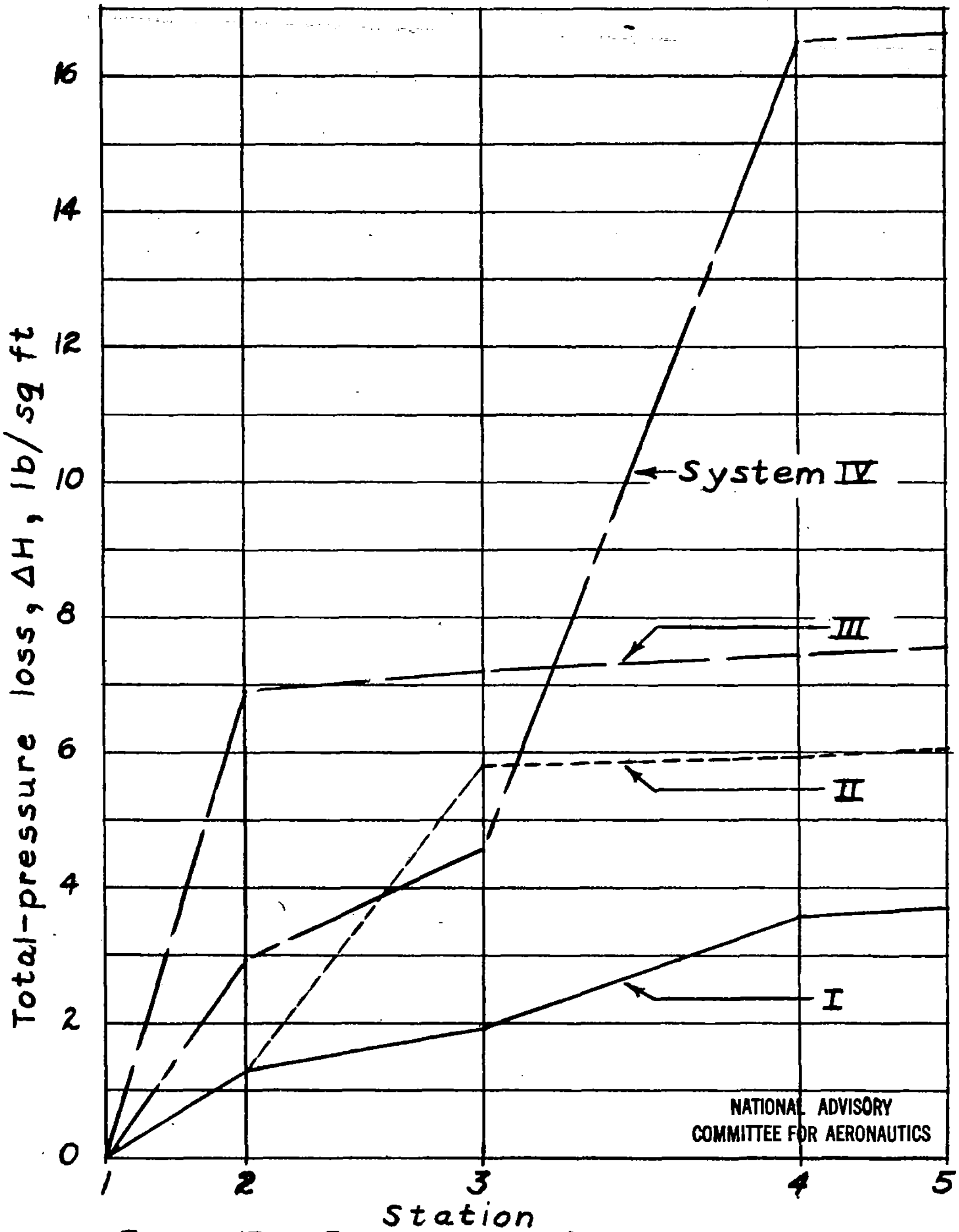


Figure 17.– Comparison of total-pressure losses through sample duct systems.

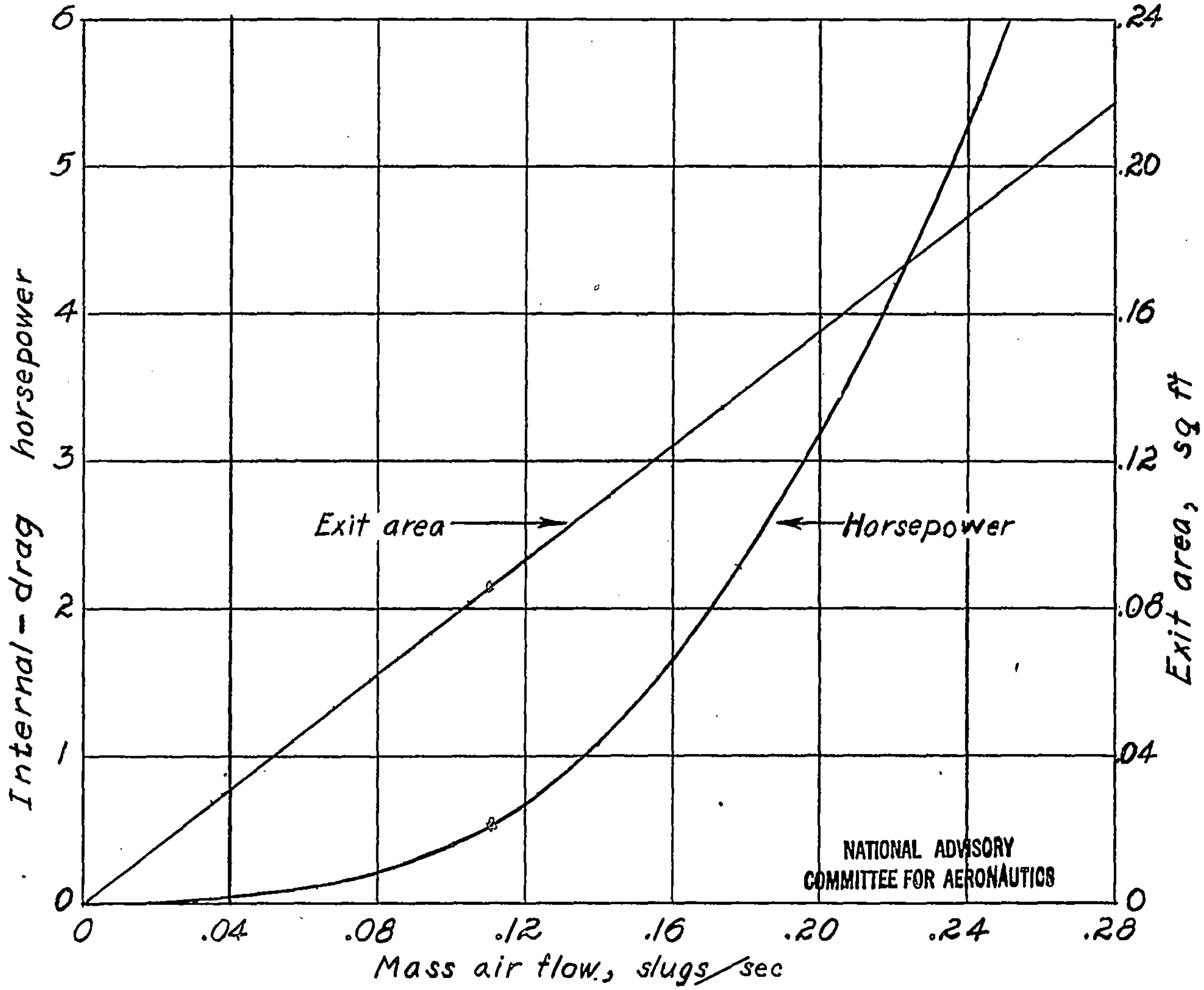


Figure 18.- Variation of exit area and internal-drag horsepower with mass flow for system I.

ERRATA No. 1

NACA ARR L4F26

DESIGN OF POWER-PLANT INSTALLATIONS
PRESSURE-LOSS CHARACTERISTICS OF DUCT COMPONENTS
By John R. Henry

June 1944

Pages 8 and 9 and figures 2, 3, and 6 have been corrected to include a calculated friction loss in the over-all loss coefficient for the bend. The corrected pages are attached to replace the corresponding pages and figures in the original version of this paper.

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velocity and the hydraulic diameter of the duct. Values of f obtained from figure 51 of reference 4 are plotted against Reynolds number in figure 1. Data in figure 13 of reference 5 agree closely with values in figure 1. Determination of the Reynolds number is facilitated by supplementary curves obtained by plotting the ratio of mass rate of flow to duct perimeter against Reynolds number for a number of air temperatures. The kenetic viscosity of the air used in constructing the supplementary curves of figure 1 was determined by Sutherland's equation as presented in reference 6.

A typical value of $\Delta H/q$ for straight aircraft ducts is $0.02 \frac{l}{D}$, which is usually inconsequential compared with other parts of the system, and the loss in sections of straight ducts is generally neglected. Long winding ducts of small diameters, such as cabin-heater ducts, are sometimes treated as straight ducts of higher than average pressure loss due to friction. The use of

$$\frac{\Delta H}{q} = 0.04 \frac{l}{D}$$

is recommended in reference 7.

<u>90° bends of constant-area rectangular cross section</u>.- Pressure-loss coefficients of 90° bends of constant-area and rectangular cross section given in figure 2 for three values of Reynolds number based on hydraulic diameter are derived from data appearing in references 5 and 8 to 12. The data of reference 5 are presented as a loss coefficient chargeable to turning which was obtained by subtracting from the measured over-all loss of the combined approach duct, bend, and tail pipe a calculated friction loss for the approach duct, bend, and tail pipe. All the bend data presented herein have been reduced to an over-all loss coefficient for the bend proper, or the data of reference 5 restored to an over-all loss by adding in the calculated friction loss of the bend. Figure 2 indicates that increasing the radius ratio beyond a value of about 2.00 yields no further reduction in loss, and that the optimum aspect ratio varies markedly with Reynolds number.

<u>90° bends of constant-area elliptical cross section</u>.- Pressure-loss characteristics of 90° bends of constant-area elliptical cross section are given in figure 3 for three values of Reynolds number. The data include circular ducts as a special case. The same general effects of radius ratio and the existence of an optimum aspect ratio are noted for the bends of constant-area elliptical cross section as well as for rectangular bends. The effects of Reynolds number are much less for bends of elliptical cross section than for bends of rectangular cross section.

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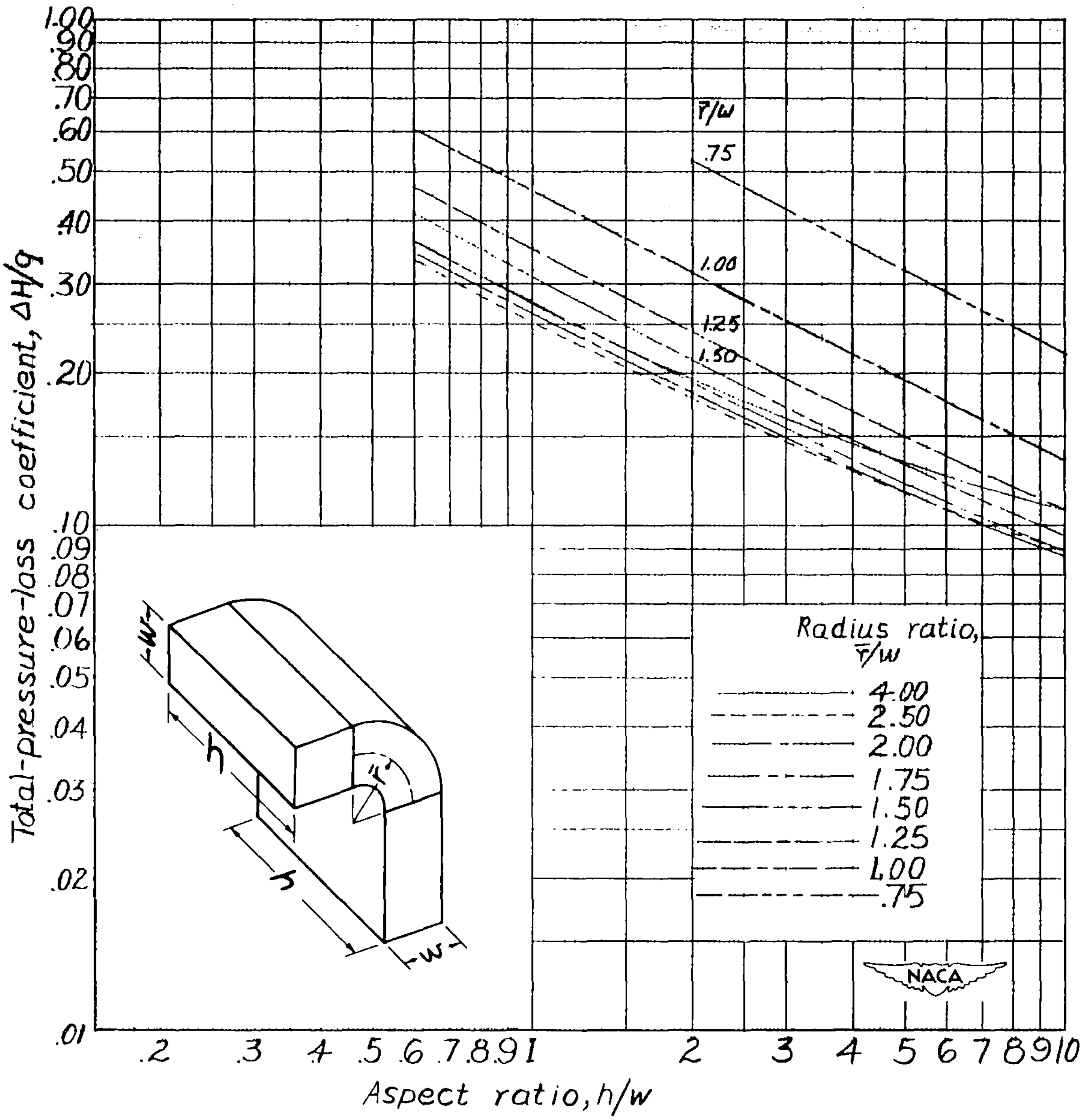
90° bends of changing area.- Significant data (derived from reference 11) concerned with the relation of area change to the loss in 90° bends of a particular geometry are shown in figure 4. In this figure the ratio of loss in a bend with changing area to that in a bend with identical inlet form but constant area is plotted against the ratio of entrance width to exit width of the nonuniform bend. Important reduction of loss in converging bends and serious increases in loss in diverging bends are noted; the loss increases are particularly serious for bends of small radius.

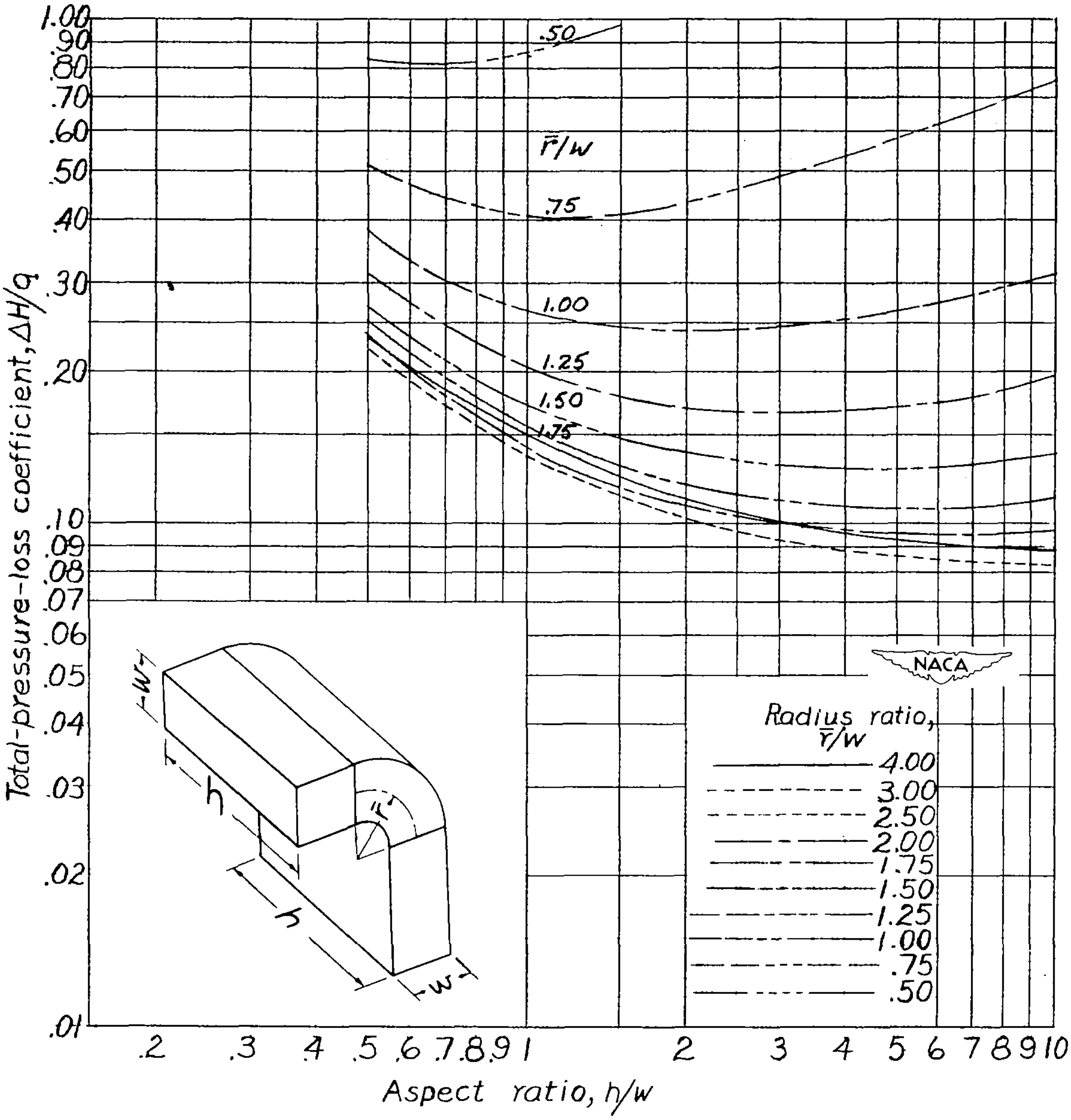
Simple bends other than 90°.- No satisfactory correlation has been made of data for variation of pressure-loss coefficient with angle of bend. Pressure loss of 45° bends can apparently vary from one-third to two-thirds the loss of a similar 90° bend, according to the test conditions.

Compound bends.- Pressure-loss coefficients for three types of compound bend (fig. 5) derived from reference 5 are shown in figure 6. Inasmuch as differences in the losses between the U-bends, Z-bends, and 90° offset bends appears from reference 5 to be small and inconsistent, the curves presented are averages of results for the three types of bend. There appears to be little variation of loss with Reynolds number. Introduction of a 5-foot spacer between the two parts of the compound bend increases the over-all loss appreciably due to the added friction loss. A comparison of the 180° bend (U-bend) data of figure 6 with the 90° bend data of figure 2 shows that the relative loss varies to a marked degree with the radius ratio and aspect ratio of the bend.

Effects of surface roughness on bend losses.- The effect of surface roughness on the losses in straight pipes has already been given by the curves of figure 1. A study of pressure-loss data for bends of angles from 30° to 90° and radius ratios from 1 to 6 (reference 11) indicates that the influence of surface roughness on the loss in bends, and presumably of other duct components in which major flow disturbances arise, is very much greater than can be attributed to the increase in skin friction at the mean velocity of flow. Analysis of the data in reference 11 suggests that the ratio of losses through two bends, identical except for surface roughness,



(a) Reynolds number, 100,000.
Figure 2.- Total-pressure-loss coefficients for rectangular 90° bends.



(b) Reynolds number, 300,000.

Figure 2.- Continued.

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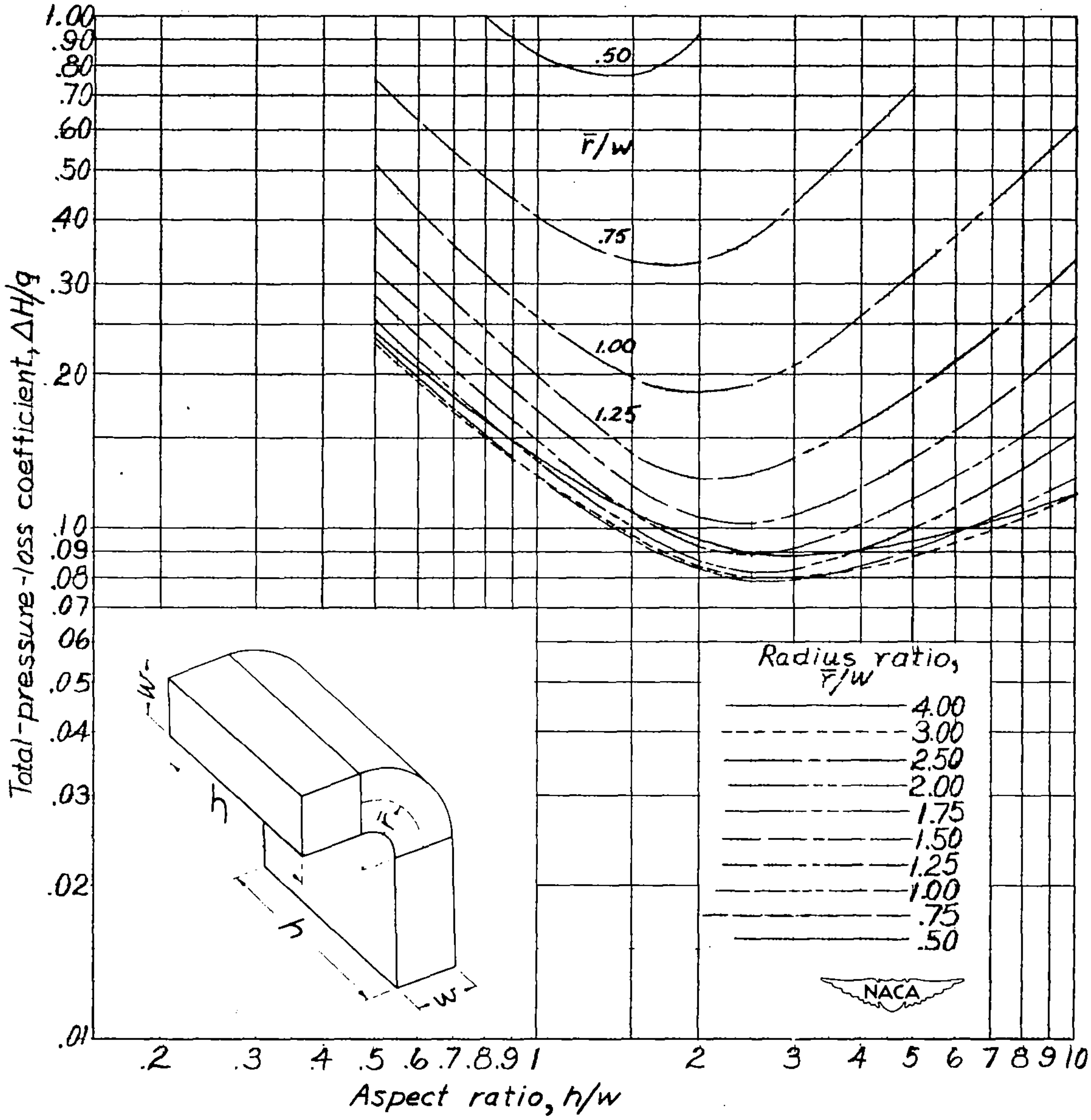
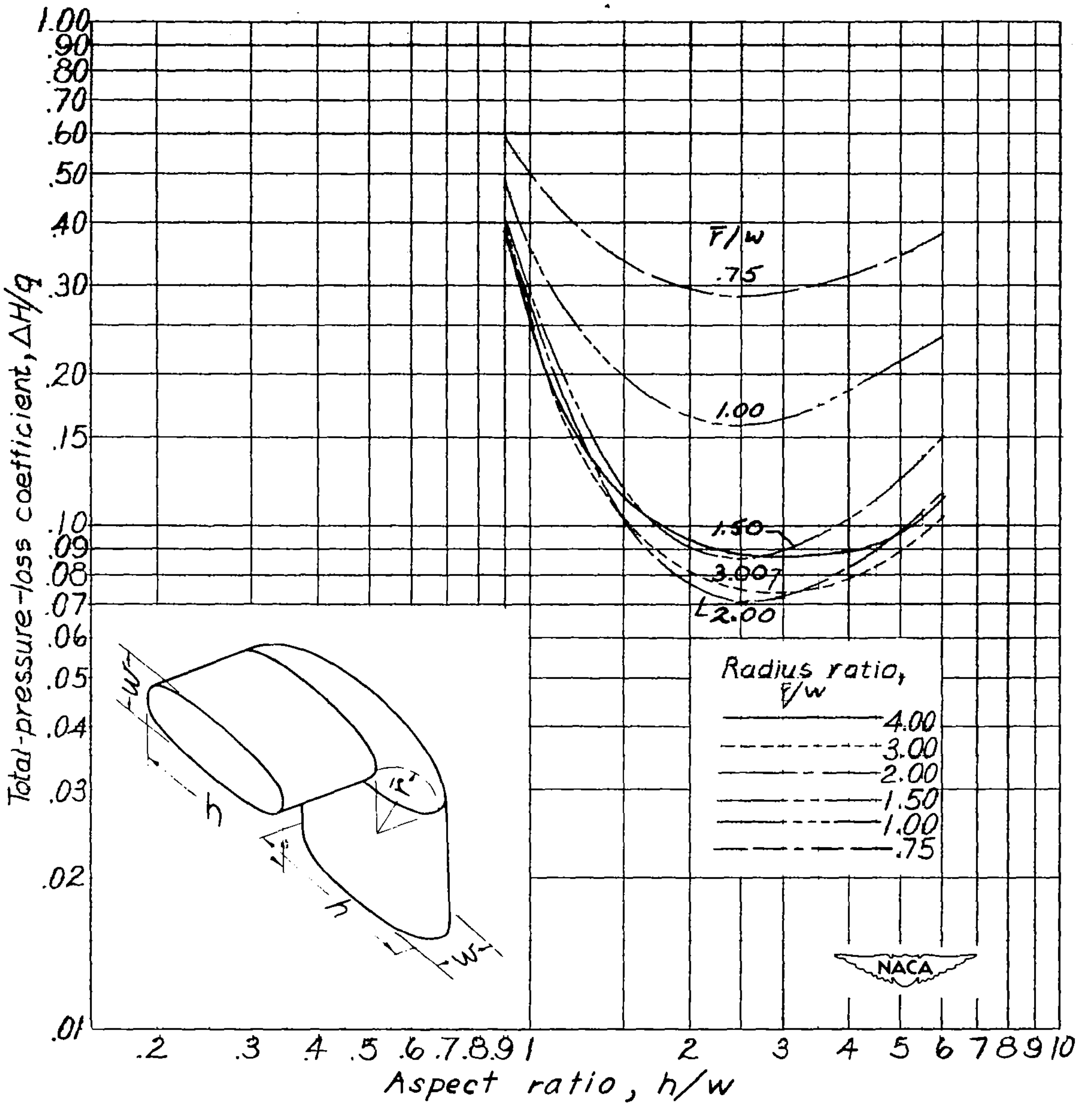
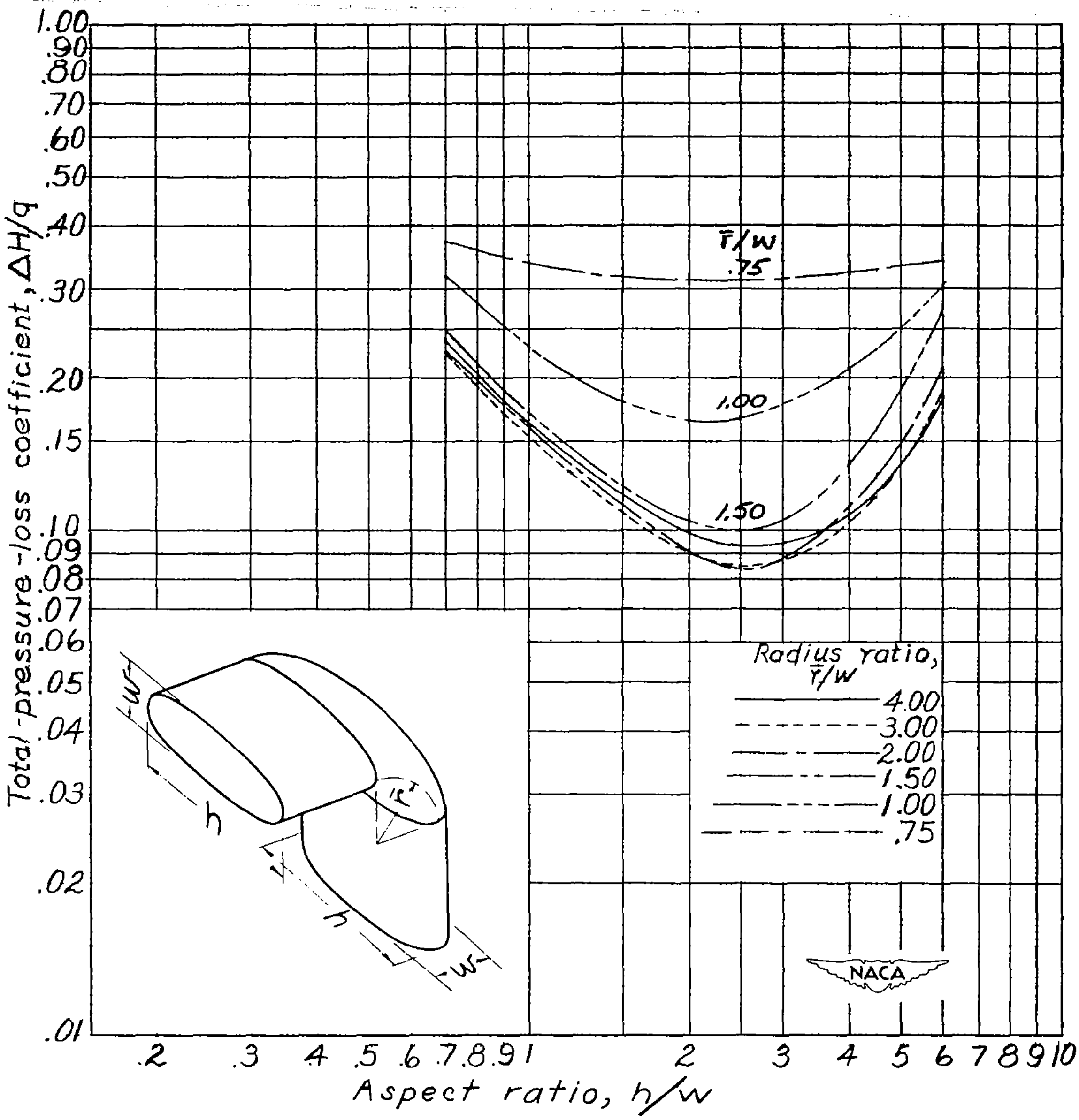


(c) Reynolds number, 600,000.
Figure 2.– Concluded.

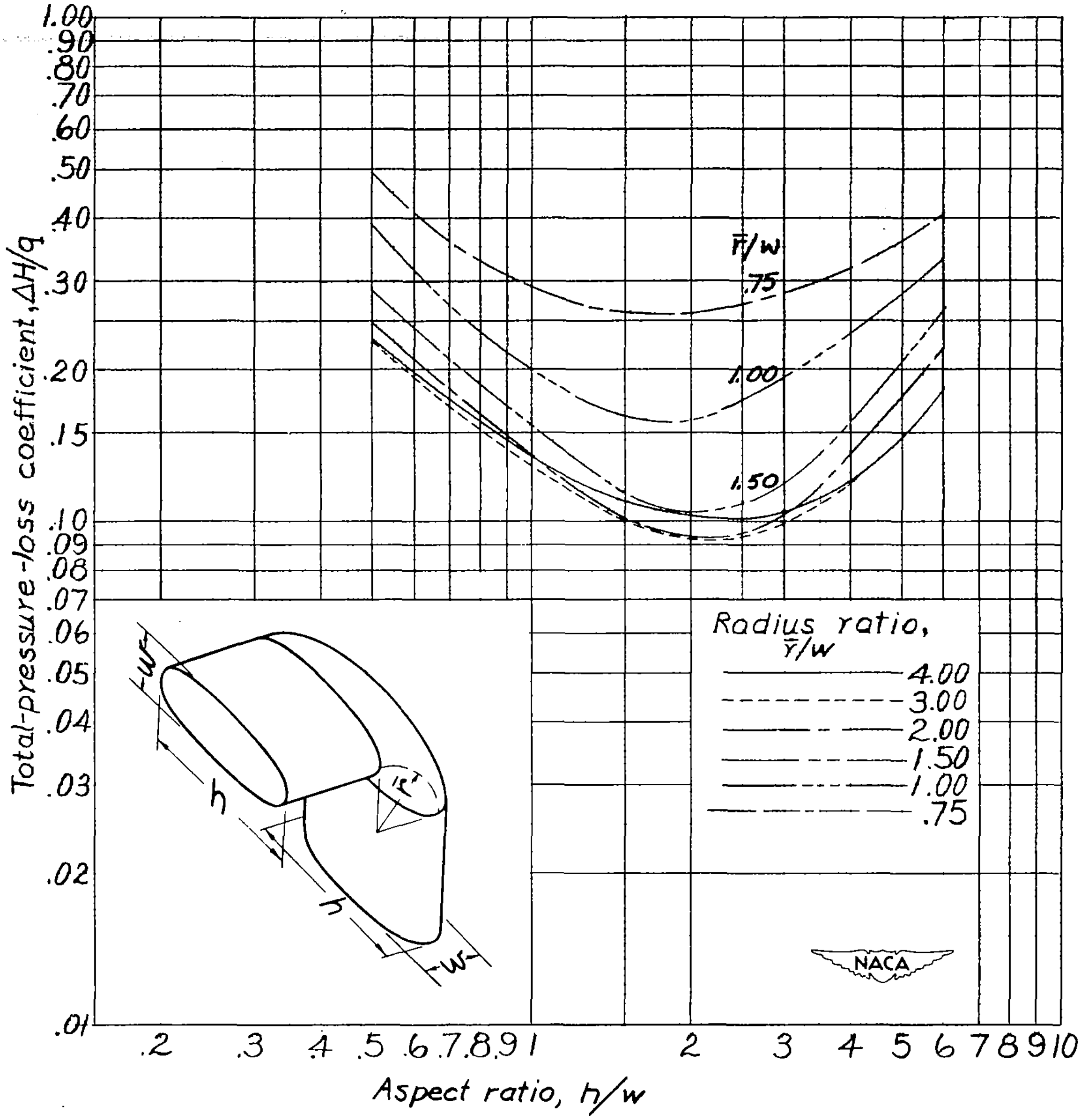


(a) Reynolds number, 150,000.

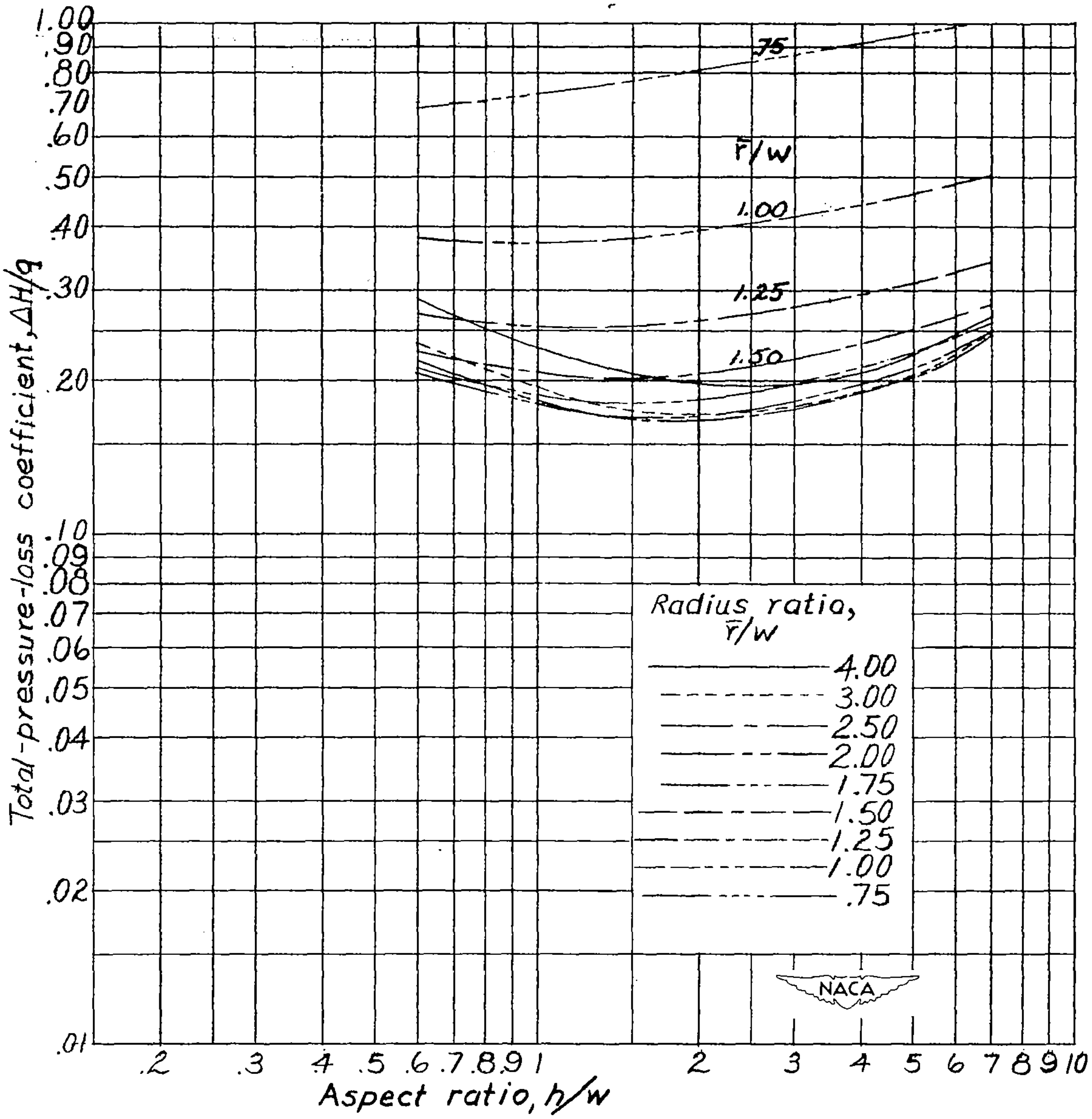
Figure 3.- Total-pressure-loss coefficients for elliptical 90° bend.



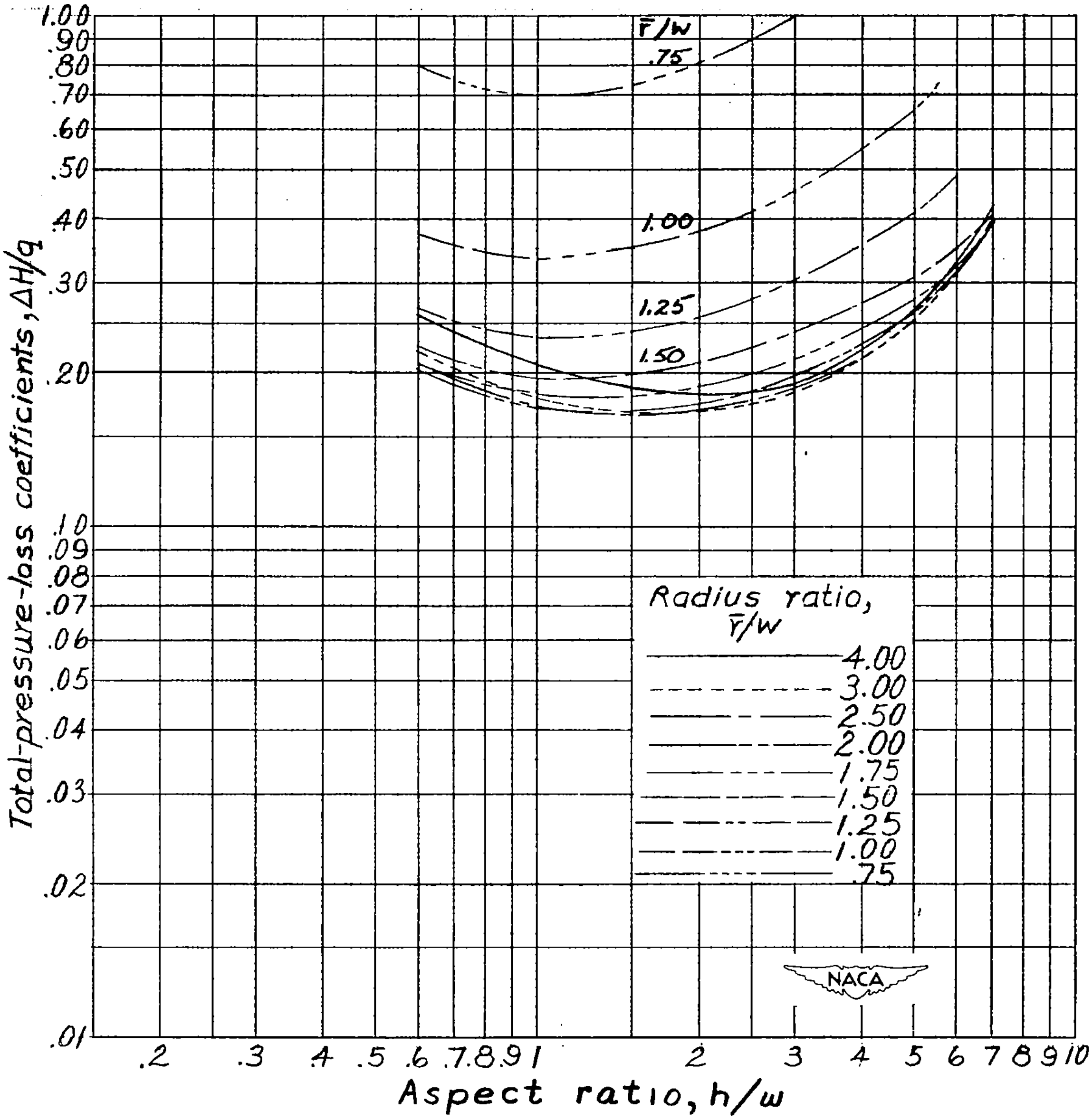
(b) Reynolds number, 300,000.
Figure 3.- Continued.



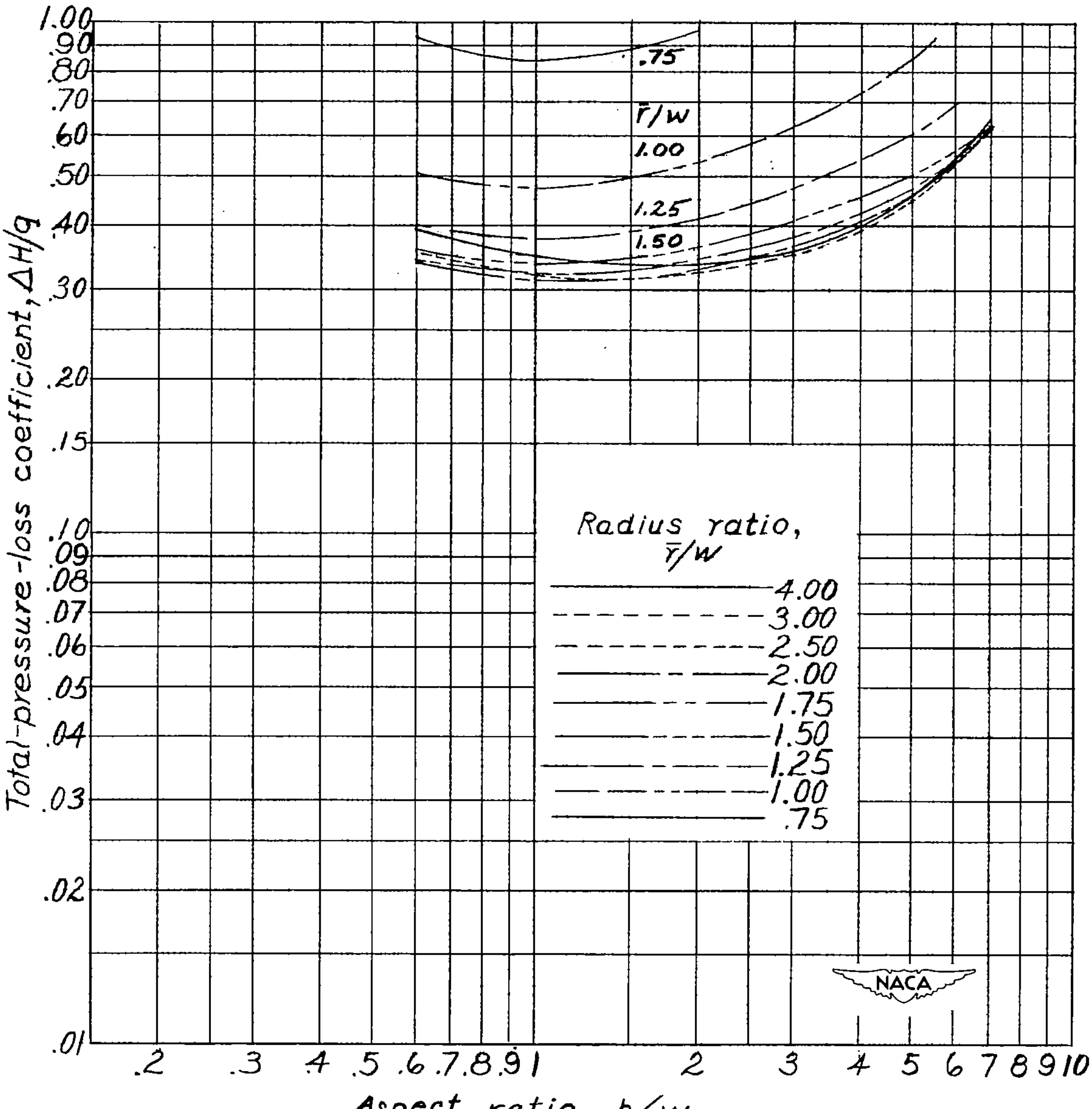
(c) Reynolds number, 600,000.
Figure 3.-Concluded.



(a) Bends without spacers; Reynolds number, 300,000.
Figure 6.- Total-pressure-loss coefficients for compound rectangular U-, Z-, and 90°- offset bends.



(b) Bends without spacers; Reynolds number, 600,000.
Figure 6.- Continued.



(c) Bends with 5-foot spacer; Reynolds number, 600,000.
Figure 6.- Concluded.